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TECHNOLOGICAL ASSESSMENT OF NEW SHIP OR ADVANCED PLATFORM CONCEPTS FOR AMPHIBIOUS FLEET OPERATIONS

Prepared for:

OFFICE OF NAVAL RESEARCH (CODE 493) WASHINGTON, D.C.

SOUTHERN CALIFORNIA LABORATORIES
OF STANFORD RESEARCH INSTITUTE

SOUTH PASADENA CALIFORNIA

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TO BASIC DOCUMENT

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In addition to the development of (1) the basic characteristics curves for each platform concept and variation, taking into consideration such factors as hull material, type of power and foil system (hydrofoil), operating height (GEM), or cross section hull shape (submarine), and (2) limit charts for each concept, integrated analyses were made to show the comparative power requirements and capital costs of alternative platforms of the same payload potential. These comparisons were developed for the range of speeds potentially feasible with each concept and for payload potentials ranging from those for the equivalent of such small naval vessels as destroyer escorts to such large vessels as tankers, cruisers, and carriers. Finally, comparisons were developed to show the probable performance characteristics and speed degradation of alternative platform concepts under adverse sea conditions.

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Organization of the Report

Following this introductory statement of background, objectives, scope, and method of approach is a brief summary of the performance characteristics that could be achieved in operational units employing one or another of the various platform concepts in the 1975-1980 time period. Section II also provides conclusions on major research or development requirements associated with each platform concept. Section III takes up in some detail the characteristics of advanced displacement hulls; Section IV takes into consideration the potential of large planing hulls; Section V provides an analysis of hydrofoil characteristics and limitations; Section VI is devoted to a discussion of ocean-going GEM's; and Section VII takes up the potential of submarines. A comparison of alternative platform concepts for fleet units of various speeds and payload potentials is presented in Section VIII. Following Section VIII are brief technical notes on reduction of hull resistances, lightweight materials, and propulsion systems, together with appropriate references.

The appendixes to this report specify in detail the assumptions and methods used in projecting the characteristics data for each platform concept. The appendixes also provide more complete sets of performance curves than are provided in the body of the report. Prepared by M. Rosenblatt & Son, Inc., Naval Architects and Marine Engineers, the appendixes are as follows: A, Advanced Displacement Hulls; B, Planing Hulls; C, Hydrofoils; D, GEM's; and E, Submarines.

DEPARTMENT OF THE NAVY
OFFICE OF NAVAL RESEARCH
WASHINGTON, D. C. 20360

IN REPLY REFER TO
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25 March 1964

From: Chief of Naval Research
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Subj: Technological Assessment of New Ship or Advanced Platform Concepts for Amphibious Fleet Operations (U); distribution of

Encl: (1) Report entitled, "Technological Assessment of New Ship or Advanced Platform Concepts for Amphibious Fleet Operations (U)"

1. Enclosure (1), final report on subject study, is forwarded for information, comment and retention. Work leading to the publication of this report was conducted by the Southern California Laboratories of Stanford Research Institute, under contract Nonr 4194(00) initiated in June 1963.

2. This study is one of six contracted for by the Office of Naval Research to assist in the determination and assessment of technological advances in various fields of relevance to amphibious assault operations of the 1975-1980 time period. The remaining studies cover weapons systems, early warning, communication, tactical deception devices and techniques, and communications deception and intelligence monitoring.

3. A prime purpose in conducting these studies is to provide technological inputs to an in-house study of amphibious assault operations currently under way by the Advanced Warfare Systems Division, Naval Analysis Group, Office of Naval Research. The in-house study is integrating these and other efforts, and formulating advanced concepts and systems in transport, combat support and command and control. Select Navy systems in support of the Landing Force are to be evaluated for feasibility, compatibility and utility in order to provide technical guidance to research and development planners.

4. The analytical effort covered by this report is focused upon advanced technology in ship or advanced platform concepts that might be developed for amphibious force operations in the 1975-1980 period.

5. The report is being released at this time because of its potential utilization by other Naval activities. Comments are elicited. Those received prior to 30 April will be given full consideration in the evaluation process of the in-house study referred to in paragraph 3.

S. Shulman

S. SHULMAN
By direction

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OFFICE OF NAVAL RESEARCH
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Marine Engineering Lab.
Annapolis, Md.

SOUTHERN CALIFORNIA LABORATORIES
OF STANFORD RESEARCH INSTITUTE
SOUTH PASADENA, CALIFORNIA

SRI

April 1964

Final Report

TECHNOLOGICAL ASSESSMENT OF NEW SHIP OR ADVANCED PLATFORM CONCEPTS FOR AMPHIBIOUS FLEET OPERATIONS

By: Irving Dow and Ephraim Kaufman

SRI Project No. ISU-4552

Prepared for:

OFFICE OF NAVAL RESEARCH (CODE 493) WASHINGTON, D.C.

Contract No. Nonr-4194(00)

Approved:

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CARLETON GREEN, GENERAL MANAGER
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PREFACE

This study was conducted for the Office of Naval Research, Washington, D.C., under Contract No. Nonr-4194(00). Mr. Absalom Simms, Assistant Director of the Advanced Warfare Systems Division (ONR Code 493), Naval Analysis Group, was the Project Officer. The study was requested by the Office of Naval Research as one of several preliminary assessments of technological developments relating to future weapons systems for amphibious warfare. These technological assessments, conducted by a number of different contractors, are intended as input studies to a larger ONR study of amphibious operations in the 1975-1980 time period.

The research was conducted by the Southern California Laboratories of Stanford Research Institute and by the Western Division of M. Rosenblatt & Son, Inc., Naval Architects and Marine Engineers, under subcontract to the Institute. Dr. Irving Dow, Manager of Research Operations, Economics Research, at the Institute's Southern California Laboratories was project leader. George Mitchell was a key member of the project team, and Dr. George Brinton also contributed to the study. Mr. Ephraim Kaufman, Manager of the Western Division, was responsible for the work conducted by M. Rosenblatt & Son, Inc. Mr. Stephen Halpern, Chief Engineer of the Western Division of that organization, made significant contributions to the project.

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I INTRODUCTION

The Office of Naval Research (ONR) has recognized that there is a continuing long-term need for the Navy and the Marine Corps to maintain and improve their capabilities for amphibious warfare. Over the past few years the Office of Naval Research has either conducted or sponsored a number of significant research projects relating to one or another facet of amphibious operations. The Office of Naval Research is now engaged in a broad-scope, in-house study of future weapons and transport systems for amphibious warfare, with the general purpose of providing guidance for research and development planning to meet the operational requirements of amphibious forces in the 1975-1980 period.

In formulating this study program, the Advanced Warfare Systems Division of ONR has been motivated by an awareness of the long range importance of the amphibious forces to the over-all military capabilities of the country and of the fact that certain technological developments afford at least a potential for application and significant improvement in amphibious warfare systems. Through its support of diverse research programs and its continuous monitoring and assessment of progress in advancement of the state-of-the-art in many technical areas, ONR has become aware that new technology could provide a means of achieving improvements in the effectiveness of command and control organization, weapons and fire support systems, and transport systems for future amphibious operations.

With these considerations in mind, the Office of Naval Research formulated an amphibious warfare systems study. As part of this over-all study program, ONR has sponsored a number of contractor studies which are intended to provide detailed assessments of future technology relating to weapons systems, VTOL/STOL transport aircraft missions, advanced ship or platform concepts, early warning systems, communications systems, and possible techniques of deception. The Institute's technological assessment of new ship or platform concepts for future amphibious operations is therefore only one of several major inputs to the larger ONR research program on amphibious warfare. It is important to note, therefore, that as an input study this report does not reach definitive conclusions on the types of ships or platforms that should be developed for future amphibious operations. Such conclusions depend not only on ship technology or on performance potentials of new ship designs per se but also on weapons technology, doctrine for the conduct of future operations, and the need for speed and deception of amphibious task force units.

It is expected that these and additional factors will be taken into account in the larger ONR study in which the results of all the input

studies will be considered and an integrated view taken of the several major components of future amphibious warfare systems. Conclusions regarding the most promising ship types, weapons, and concepts of employment will, of course, be dependent on the postulation of alternative amphibious warfare systems, described in terms of major components, and on analysis of comparative system costs and potential effectiveness in meeting operational requirements for various types of missions that the amphibious forces might be called on to conduct.

Study Objectives

The over-all objective of this study is to provide, as an input to the larger ONR study of future weapons systems for amphibious warfare, an assessment of the technical feasibility and probable performance characteristics of advanced ship or new platform concepts that might be developed as fleet units for amphibious task force operations in the 1975-1980 period. Specific objectives are:

1. To project the potential performance characteristics of advanced ship or new platform concepts, treating parametrically such factors as over-all size, payload, power, speed, and range.

2. To compare the maximum design speed and payload potentials and the technical feasibility of the various alternative concepts, identifying the most critical technical problem areas.

3. To assess the probable differences in capital costs and the comparative operational capabilities or limitations inherent in each platform concept.

Scope and Method of Approach

The scope of the research includes consideration of a number of advanced design concepts that have been quite widely discussed as possible successors to present-day ships to meet future naval requirements. In addition to advanced displacement hulls that could afford significant increases in speed (anticipating possible improvements in propulsion systems, hull materials, and hull design or other means of reducing wave resistance) consideration has been given to the following platform

concepts: hydrofoil ships, planing hulls, ground effect machines (air cushion ships), and submerged vessels. Specifically excluded from the scope of the research were fully airborne platforms (such as the seaplane) or space platforms, both of which have been suggested as possibly having some potential naval applications as replacements for present ship types.

The present-day amphibious task force is composed of (1) amphibious shipping units, such as the LSD, LPD, LPH, LST, AKA, APA, and AGC, which transport landing force units, equipments, supplies, and the means of projecting the landing force ashore, and (2) combatant ship units, such as DD, DLG, CL, CA, CVS, and CVA, which provide anti-submarine and anti-air warfare protection for the transport group of the amphibious task force, strike forces for operations in advance of or in conjunction with the assault in the objective area, and supporting fire for the landing forces. In short, the present-day amphibious task force is composed of ships of a broad range of sizes and may include virtually all types of major fleet units. The particular types of fleet units that will be required in the 1975-1980 period will, as indicated earlier, depend on such factors as the speed of the ship or platform used, the technique employed to afford protection of transport groups en route and in the objective area, and the weapons systems being used. It is evident, however, even at this juncture that the various missions of the different groups and elements of the amphibious task force will require ships or platforms of differing characteristics and sizes.

Historically, ships have had an active service life of twenty years or more. However, future technological change in both ships and weapons systems could reduce significantly the useful life of major fleet units. As radically new ship or platform design concepts are introduced, the inventory of ships available will become a mix of old and new ship types, a situation which could create difficult planning and operational problems. Adoption of ocean-going GEM's to replace present types of assault shipping could also require the adoption of high-speed combatant type vessels or could result in a change in amphibious doctrine. Certain advanced vessel or platform concepts may well have a significant influence on the types of weapons systems that should be developed for the amphibious task force and amphibious operations of the future. Conversely, the development of new weapons systems will influence ship design requirements. In the development of future weapons systems concepts, there is a need, therefore, to consider the types and sizes of ships or platforms that might be feasible from a technical and operational standpoint.

In view of the foregoing, it was considered essential in this study to consider the possible characteristics and the feasibility of each alternative platform concept--for example, advanced displacement hull,

planing hull, hydrofoil, GEM--over the full range of possible sizes that might be required for various units of the amphibious task force of the future. Also, it was considered essential to develop some means of comparing alternative platform concepts on an equivalent basis. Inasmuch as each of the various platforms offers essentially different speed characteristics, speed as such would not provide an <u>equivalent</u> basis for comparing alternative concepts. Accordingly, equivalent payload potential was adopted as the basis for comparing alternative platform concepts. While "payload" is not a normal measure of the size or capacity of a naval vessel, it is useful here not only in comparing platforms on an equivalent basis but also in providing a perspective as to the size of ship or platform required for particular missions within the amphibious task force. Thus, the payload equivalent of a present-type DD, CG, AKA, or any other naval ship can be used as a basis for considering what the performance potential of an equivalent-size hydrofoil, ground effect machine (GEM), or submarine might be.

Payload equivalent is defined in Fig. 1 as that part of the total full-load displacement of a vessel that is available for the installation and accommodation of armament, communications and electronics (associated with weapons systems operation in contrast to operations of the ship), ammunition, crew, and stores. In the equation shown in the figure it may be seen that payload is that part of total full-load displacement not accounted for by weight of the hull, propulsion system, electric plant, communications and electronics (for ship operations),[1] auxiliary systems, outfitting, and fuel. As the equation implies, payload potential in tons is indicated for a particular range because fuel weight must be specified. The payload equivalents of various existing classes of naval vessels shown in the figure are based on actual weight data for each BuShips weight category (W_1 through W_6) and fuel weight (W_F), assuming full capacity of fuel for each class of vessel.

It may be seen from the payload equivalents of the various classes of vessels shown in Fig. 1 that vessels or platforms with payloads of from perhaps 250 tons (DE-1033 = 233 tons) to several thousand tons are of potential interest in amphibious operations. One of the major tasks in this study was to develop for each alternative platform concept a basic

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1. Since communications and electronics systems are essential to both operation of the ship (as a platform for the payload it is carrying) and operation of the weapons systems (constituting in combatant ships the primary portion of payload) this weight category (W_4) has arbitrarily been allocated one-half to payload and one-half to the ship itself.

PAYLOAD EQUIVALENT $(\Delta_p) = \Delta - (W_1 + W_2 + W_3 + \frac{1}{2}W_4 + W_5 + W_6 + W_F)$

WHERE

- Δ = FULL LOAD DISPLACEMENT
- W_1 = WEIGHT OF HULL STRUCTURE
- W_2 = WEIGHT OF PROPULSION SYSTEM
- W_3 = WEIGHT OF ELECTRIC PLANT
- W_4 = WEIGHT OF COMMUNICATIONS & CONTROL SYSTEMS
- W_5 = WEIGHT OF AUXILIARY SYSTEMS
- W_6 = WEIGHT OF OUTFITTING & FURNISHINGS
- W_F = WEIGHT OF FUEL

PAYLOAD EQUIVALENT OF SELECTED NAVAL VESSELS

SHIP NAME	FULL LOAD DISPLACEMENT LONG TONS	EQUIVALENT PAYLOAD LONG TONS	PAYLOAD PERCENTAGE	SHIP NAME	FULL LOAD DISPLACEMENT (LONG TONS)	EQUIVALENT PAYLOAD (LONG TONS)	PAYLOAD PERCENTAGE
CVA N-65	87 655	18 322	21	LPH-2	18 004	1,456	8
CL-106	14 464	1 649	11	LPD-1	13 654	3,576	26
CG N-9	16 326	3 011	18	LSD-28	12 366	2 978	24
DD-931	3 919	735	19	LST-1173	7,356	2,350	32
DLG N-25	8 515	1 366	16	AKA-112	15 970	4 226	26
DE-1033	1 694	233	14	APA-248	16,576	3,429	21
SS N-593	3 750	652	17	AE-23	17,510	7,615	45
SSB N-608	6 946	2 119	31	AF-58	15,145	5,387	36
				AO 143	37,693	21,307	57

FIG. 1 DEFINITION OF PAYLOAD EQUIVALENT AS A BASIS FOR COMPARING ALTERNATIVE PLATFORM CONCEPTS

set of curves showing fundamental relationships among such characteristics as payload, total displacement, speed, horsepower, and range at maximum design speed. These curves show characteristic data on platforms, with payload equivalents ranging from those for the smallest to the largest size ships or platforms of interest or the largest size platforms feasible for a particular platform concept. Certain of the concepts, as will be seen, are limited clearly to small-size platforms.

At the outset of the work, consideration was given to the types of ships that might be involved in amphibious operations and to the organization and composition of the amphibious task force. In this preliminary step, discussions were held with a number of personnel in the amphibious fleets, and a review was made of current doctrine and directives regarding the conduct of amphibious operations. Discussions on how the amphibious task force would be organized, how operations would be conducted, and what types of transport and combatant vessels would be required if one or another of the high-speed or submersible platform concepts were adopted were inconclusive. It was demonstrated that such questions could be answered or analyzed only within the context of a statement of task force objectives, the enemy situation and the threat to the task force, the types of weapons employed, and the pattern of strategic deployment or disposition of U.S. forces. From these discussions it was evident that a major requirement in this study of future platform concepts was development of the basic curves from which the critical characteristics of a platform of any given size (payload equivalent) or speed could be determined. By use of these curves, the most suitable platform concepts for use in conjunction with particular weapons systems, deployment patterns, or types of situations could be selected. This, of course, is expected to be within the scope of the larger ONR study but beyond the scope of this study.

To establish a basis for projecting the potential performance characteristics and design limitations on the speed and payload potential of alternative platform concepts, it was essential to examine carefully the possible timing of the various steps or phases required in a major program for the development and delivery of a significant number of ships or platforms of a new type to the operating forces in the 1975–1980 period. The time required for such steps as feasibility studies, experimental model testing, preliminary design, contract design, bidding, construction, sea trials, and prototype delivery, followed by series production of fleet units, is illustrated in Fig. 2. In the figure it is seen that approximately nine years would elapse from the initiation of preliminary design to the acceptance of the prototype platform, and that series production and the buildup to significant numbers in the fleet would take several more years.

The schedule presented in Fig. 2 is generalized, and it is clear that the actual development schedule would depend on (1) the particular platform concept being developed, (2) the priority assigned to the program, including the resources made available, and (3) the success in technological development required for actual construction and delivery of prototype and production models. While the figure is generalized, the conclusion is clear: projections of the potential performance characteristics of platforms feasible for operational use in the 1975-1980 period must be based on an assessment of what is technically feasible within the present state-of-the-art or will be feasible within the next three to four years at the latest. In other words, the projections of future platform characteristics cannot be based on what might be technically feasible in 1975. Technical guidelines have to be established and major technological problems solved in the preliminary design stage which, as shown in the figure as an initial step in the over-all development program, may take as long as three years. This characteristically long lead time in the design, construction, and delivery of major fleet units has been taken into account in the technological projections and statements of performance potentials of platforms that could be in operational use in the 1975-1980 time period.

Technological projections and parametric charts to show probable performance characteristics of each alternative platform concept have been developed utilizing detailed technical studies available on advanced hull forms, means of reduction of wave resistance, new hull materials, and highly advanced and lightweight propulsion systems. Technological studies undertaken for such agencies as ONR, BuShips, and Maritime Administration, as well as various other governmental and industrial organizations, have been used in investigating future technology and extrapolating design data. The present and possible near-future state-of-the-art has been examined with respect to critical technical problems affecting the various platform concepts, and significant technical limitations have been identified. Major research and development problems are thus indicated for each platform concept. The project team has not undertaken any basic research to advance the state-of-the-art relating to the technological feasibility of particular types of ships or platforms.

Professional judgment has been important in extrapolating existing design and experimental data and in interpreting theoretical studies relating to probable performance characteristics. Also, professional judgment has been important in assessing the likelihood or feasibility of achieving advances in the state-of-the-art and in making realistic assumptions regarding appropriate design parameters, such as the displacement-length ratio for the high-speed advanced displacement hull, the achievable lift-to-drag ratio for large hydrofoil vessels, and the

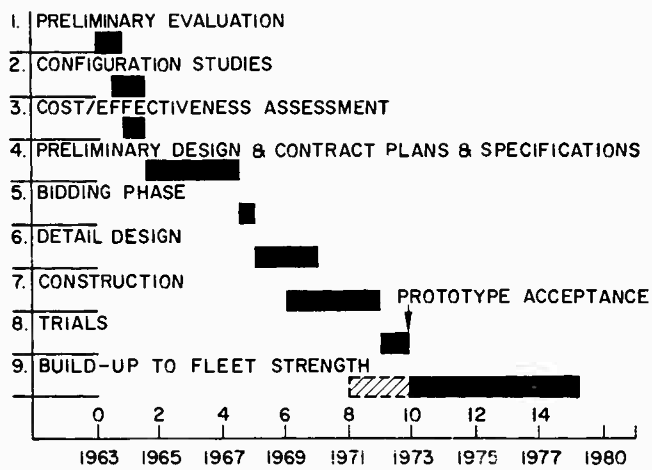

FIG. 2 PROBABLE DEVELOPMENT SCHEDULE FOR ADVANCED PLATFORM CONCEPTS

lift-base area ratio for the GEM. Values assumed for such ratios are, of course, critical to determination of performance potentials and the comparative feasibility of alternative platforms of various payload and speed characteristics.

Limit charts have been developed to show for each platform concept the maximum feasible payload potential as a function of design speed under specified conditions or assumptions regarding range, type of propulsion plant, hull materials, and so on. In some cases, in increasing the design speed of a vehicle with a fixed payload, a point is reached at which further increases in installed horsepower and fuel actually would require an increase in the vehicle size and would result in reduction of the speed in order to attain the same payload. In these cases there is a physical limit to the feasible payload and speed that can be achieved. In other cases there is no clear physical limit--further increases in installed power can provide further increases in speed--and, ignoring costs, it becomes a matter of judgment as to how much power could be installed in a given platform and how much power could in fact be converted to effective thrust. The limit on payload or speed or both may also be a matter of judgment relative to the maximum-size hull structure it is feasible to construct. This is indeed a consideration in the cases of the hydrofoil and the GEM.

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II SUMMARY

A major objective of this research has been to project the potential performance characteristics of advanced ship or new platform concepts that might be considered for amphibious operations in the 1975-1980 time period. Technical assessments have been made of the maximum operating speeds that might be achieved by different platforms of size or payload capacities that could be required for fleet units in the amphibious task force of the future. A significant fact recognized at the onset of the research is that there is only a limited amount of time available for formulation of a major development and construction program if new ship or platform designs are to be delivered to the operational forces in significant numbers in the 1975-1980 period. Accordingly, it was important to give careful consideration to the possible timing and the sequence of the various phases of such a program, such as the experimental model testing phase, the preliminary design phase, and the detail phase, in these assessments of the technical feasibility and performance potential that would be achieved in high-speed displacement hulls, planing hulls, hydrofoils, ground effect machines, and submarines within the time period of interest.

The fundamental reason for consideration of these particular platform concepts is that each offers the possibility of significantly greater speed than is characteristic of present naval vessels. Various of the alternatives also offer other potential advantages. For example, the submarine offers the potential for deception; the GEM offers the potential for beaching across shallow water approaches and for off loading landing forces directly on the shore; and the hydrofoil offers good performance under adverse sea conditions. Moreover, certain of the alternatives are inherently less costly than others of the same payload potential and speed. Nevertheless, the basic interest in these new ship or platform concepts is the possibility of achieving comparatively high-speed capabilities for ocean-going vessels.

Table I is a summary table showing, for each alternative platform concept, the maximum speed projected to be feasible in operational designs. The speed potential of each concept is indicated for platforms with payload equivalents ranging from 250 tons to 10,000 tons, and both nuclear and non-nuclear propulsion systems are considered.

The most significant single point to be noted in the table is the extremely limited payload potential of both planing hulls and hydrofoils. In each case the maximum feasible payload equivalent is 1,250 tons, and the maximum speed at this payload is 30 knots, which is less than that of present displacement hulls. At a 750-ton payload equivalent, the

Table I

MAXIMUM SPEED POTENTIAL, ALTERNATIVE PLATFORM CONCEPTS
(Payload Equivalents, 250-10,000 Tons)

PLATFORM CONCEPT / PAYLOAD EQUIVALENT	ADVANCED DISPLACEMENT HULLS				PLANING HULLS (ALUMINUM)		HYDROFOILS (3,000Δ, max)			
	STEEL		ALUMINUM				SUBCAVITATING		SUPERCAVITATING	
	STEAM TURBINE[1]	NUCLEAR REACTOR	GAS TURBINE[1]	NUCLEAR REACTOR	GAS TURBINE[2]	NUCLEAR REACTOR	GAS TURBINE[2]	NUCLEAR REACTOR	GAS TURBINE[2]	NUCLEAR REACTOR
DE (250 T)	50	62	40	41	80	68	60	51	83	X
DD (750 T)	49	60	47	44	80	41	55	34	X	X
DLG (1250 T)	48	58	50	46	30	X	30	X	X	X
LST (2000 T)	47	55	55	52	X	X	X	X	X	X
LPD, APA (3500 T)	46	51	58	62	X	X	X	X	X	X
AKA (5000 T)	45	49	57	57	X	X	X	X	X	X
10,000 T	43	44	49	50	X	X	X	X	X	X

PLATFORM CONCEPT / PAYLOAD EQUIVALENT	GEM (3,000Δ_t max)				GEM (10,000Δ_t max)				SUBMARINES (DRY CARGO, RECTANGULAR)	
	H=10'		H=20'		H=10'		H=20'			
	GAS TURBINE[1]	NUCLEAR REACTOR	GAS TURBINE[1]	NUCLEAR REACTOR	GAS TURBINE[1]	NUCLEAR REACTOR	GAS TURBINE[1]	NUCLEAR REACTOR	25,000Δ_t MAX	50,000Δ_t MAX
DE (250 T)	100	100	X	65	100	100	100	100	45	45
DD (750 T)	X	60	X	X	100	100	100	100	45	45
DLG (1250 T)	X	X	X	X	100	100	100	100	45	45
LST (2000 T)	X	X	X	X	100	100	X	100	45	45
LPD, APA (3500 T)	X	X	X	X	60	100	X	95	42	45
AKA (5000 T)	X	X	X	X	X	X	X	X	38	45
10,000 T	X	X	X	X	X	X	X	X	X	42

X NOT FEASIBLE WITHIN CONSTRAINTS ESTABLISHED
1. 2,000-MILE RANGE
2. 500-MILE RANGE

planing hull might achieve an 80-knot speed and the hydrofoil a 55-knot speed, assuming gas turbine propulsion and a range of only 500 miles for both the planing hull and the hydrofoil. With nuclear propulsion, a planing hull of 750-ton payload would have a maximum speed of 41 knots and the 750-ton hydrofoil a maximum speed of 34 knots.

It is observed that the high-speed potential of a supercavitating hydrofoil (80-100 knots) is achievable only with a very small payload equivalent. A 250-ton supercavitating hydrofoil would have a maximum speed of 83 knots and a range of only 500 miles. Larger payloads for supercavitating hydrofoils are not feasible within the projected state-of-the-art relating to weights of structure, foils, and power plant.

The ground effect machine (GEM) offers high speed and large payload equivalents only if very large GEM structures can be designed, as indicated in the table. If the maximum-size GEM feasible in the time period of interest is 3,000 tons, the GEM, like the hydrofoil and the planing hull, offers the potential for only DE and DD payload equivalents. Under the assumption that GEM's with a total gross weight of 10,000 tons can be constructed, payload equivalents up to 3,500 tons and speeds of 100 knots can be achieved if lightweight nuclear propulsion is used.

The high-speed displacement hulls offer the potential for maximum speeds of approximately 45 to 60 knots, depending on hull construction and type of propulsion. The aluminum hull high-speed displacement vessel offers significant speed advantages over steel hull vessels at higher payloads. At lower payloads the theoretically achievable speed of the aluminum hull vessel is less than that for steel hulls because the smaller, lighter weight displacement hulls reach a critical upper speed level where there is danger of swamping. Because of its lighter weight and shorter over-all length. the aluminum hull encounters this critical point at lower speeds than do steel hull vessels of the same payload potential.

The maximum speed achievable with the submarine is shown to be about 45 knots. If very large submarines of 50,000 tons displacement or more can be constructed, as seems feasible, this speed can be achieved for virtually the full range of payload equivalents shown.

In brief, if only moderately faster vessels are required, the most attractive alternative would appear to be the high-speed aluminum displacement hull, probably with nuclear propulsion. Such vessels would offer speeds of from 45 to 60 knots, depending on payload equivalent. The high-speed aluminum hull powered by a gas turbine power plant would offer about the same speed potential but would have a range limit of

2,000 miles. The capital cost of the nuclear vessel would be from 20 to 25 percent higher. Operating costs have not been compared. Submarines also offer modest speed increases for vessels of virtually any payload potential, but costs are very high. For higher speeds (60 to 100 knots) it is clear that the GEM is the only alternative that offers the potential for high payload capacities. It is clear also that nuclear propulsion would be advantageous, if not mandatory, for the larger GEM.

It is important to note that each of the various alternative platform concepts has a different performance characteristic under adverse sea conditions. Operation of planing hulls at design speeds would be feasible only under the most favorable sea conditions. The hydrofoil, by contrast, can be designed to operate at design speeds over extremely rough waters. The performance of ocean-going GEM's over higher sea states is not well understood, but it is clear that ability to operate over rough seas will depend to a considerable extent on operating height. With higher operating heights the GEM pays severe penalties in power requirements and costs. With respect to displacement hulls, there is evidence that high-speed displacement hulls may be characterized by very good rough sea capability, having sufficient power to reach and maintain "supercritical" speeds in heavy seas. The submarine, of course, is generally insensitive to winds and waves and can maintain speeds despite the sea conditions if operating at sufficient depth.

Achievement of projected performance potentials in operational units employing one or another platform concept would be dependent on significant advances in the states-of-the-art in several areas. The implicit assumption in the projections shown is that these advances could be successfully realized during the preliminary design phase of a long-term development and construction program. However, these necessary advances in the states-of-the-art will probably not be forthcoming unless specific developmental requirements are established and research and developmental programs initiated, as appropriate.

Probably the most significant developmental need is for lightweight nuclear propulsion systems. The maximum performance potential of both high-speed aluminum hull displacement vessels and large ground effect machines is dependent on the utilization of lightweight nuclear power plants. In the case of the GEM, this is particularly important. To some extent the development of lighter-weight nuclear plants is also important to achieving the projected potential of large, high-speed submarines. Power plant weights assumed as the bases of projections in this study can be achieved for large plants with high horsepower ratings. Significant advantages would result if further substantial weight reductions could be achieved, especially for smaller plants where the relative weight of

shielding required is prohibitive. For smaller high-speed platforms, nuclear propulsion is in some cases totally infeasible because of shielding weight and in other cases results in very poor payload capacity relative to the over-all size and weight of the platform. It may well be that basic research into possible new techniques for shielding or means of avoiding the necessity of heavy shielding are in order.

A second major research and developmental need is in the area of high-energy thrust devices. The high-speed potentials described here, particularly for larger platforms, are all dependent on effective utilization of power plants of extraordinarily large capacity as compared with the horsepower of present propulsion systems. Attention should be drawn to several aspects of this over-all problem of effectively utilizing high-capacity power plants. There are limitations on the amount of power that can be applied to a single shaft. Further development of high-speed supercavitating water propellers is an apparent requirement for high-speed displacement hulls, planing hulls, larger hydrofoils, or any platform concept employing a water-propeller propulsion system. Alternative propulsion systems warrant continued investigation and development; however, the outlook is not promising in terms of the propulsive efficiencies that appear realizable to date, as in the water-jet system. The capacity of angle-jointed power plant-propeller shafting systems may well be a constraint on speed and payload achievable in large hydrofoils and perhaps GEM's. The assumption has been made that it would be technically feasible to apply up to 50,000 horsepower per propeller for hydrofoils; this is more than twice the power of existing angle-jointed power transmission systems. Alternative approaches to right-angle power transmission may be required for very-high-capacity power applications.

The projections of the technical limits of speed and size and the performance potentials of the GEM are based largely on theoretical studies and the extrapolation of actual design and operating or test data on very small ground effect machines. There is a critical requirement for immediate construction and testing of a relatively large, experimental ocean-going GEM, if the projected potential of large GEM's is to be realized in the time period of concern here. Principal technical requirements relate to: identification and solution of control and maneuvering problems of large GEM's, achievable performance characteristics over undulating and rough seas at high speed, design operating heights required as a function of GEM size and probable sea conditions, and establishment of specific design criteria for structures as a basis for completion of preliminary and detail design of large GEM's.

The projected feasibility of high-speed displacement hulls has been based on designs employing fine hull lines and very large amounts of power to overcome resistance. Possible alternative means of minimizing hull resistance have been examined briefly (including pumping off the boundary layer of water and use of skin coatings to reduce resistance), but the findings thus far do not indicate significant promise for new design in the time period of interest. Nevertheless, it is submitted that continued research into possible means of significantly reducing the resistances of high-speed displacement hulls is warranted. The potential payoff could be very great in making high-speed displacement hulls feasible at lower costs.

As a final commentary it is observed that it is technically feasible to build high-speed ocean-going ships or platforms, but that the costs are extremely high. Comparative capital costs have been investigated; there is now a need to develop detailed estimates of annual operating costs, including prorated investment costs, for various types of high-speed ships or platforms. Moreover, as a basis for decisions regarding possible developmental and construction programs for new high-speed platforms for amphibious operations, there is a fundamental need to investigate the requirement for speed or the possible effectiveness of using platforms of various higher speeds for future amphibious operations. Also, there is a need to assess the requirement for, and the potential advantages or effectiveness of, submarine platforms having speeds of 45 to 50 knots for amphibious operations. Analysis of the comparative costs and the operational effectiveness of, or advantages obtained by, these alternative platform concepts would then provide the basis for establishing and planning high-speed ship or platform development programs.

III ADVANCED DISPLACEMENT HULLS

The feasibility of achieving significant increases in the speed of displacement hull ships has been investigated. Present-day ships are generally limited to top design speeds of 30 to 35 knots. Power requirements and costs increase sharply with further increases in speed, and the economics of ship construction and operation have tended to preclude the design or construction of large ocean-going ships with speeds greater than 35 knots. In fact, as a point of interest it is noted that the design speed of newly constructed or presently programmed amphibious ships (LPH, LPD, AKA) is, for economic reasons, limited to about 20 knots. This is far slower than present technology would permit.

It is recognized that significant advances in the speed of ocean-going vessels will be costly, whether the design concept employed is a high-speed displacement hull, a planing hull, a hydrofoil, or some other concept. One of the purposes of this study is to determine the most feasible means of achieving high-speed capabilities. The economic justification or operational requirement for high-speed ocean-going platforms has not been investigated. In this section the feasibility of achieving significant increases in the speed of displacement hull ships is discussed, considering both steel and aluminum hull ships and steam, gas turbine, and nuclear propulsion.

Hull Form and Power Requirements

As design speed is increased, the power required for a vessel of a given size and total displacement increases sharply because of extraordinary increases in hull resistance (see horsepower curves, Figs. A-1 and A-2, Appendix A). Compounding the growth in power required for higher design speeds is the increase in propulsion plant weight and fuel requirements associated with higher horsepower and, for a vessel of a given payload, the necessary increase in total displacement tonnage. If total displacement is held constant, an increase in the design speed results in a decrease in payload potential. As a consequence, in investigating the feasibility of designing displacement hull ships capable of very high speeds (well above the present 30-35 knots characteristic of the faster naval vessels), a number of approaches were examined. Consideration was given to means of significantly reducing hull resistances (thereby reducing power requirements and effecting savings in weight of propulsion plant, hull structure, and fuel), including variations in hull form, such as the bulbous bow, techniques for sucking off the boundary layer of water along the hull, and use of a hull coating to absorb the energy in perturbations

in the water. Also, the use of lightweight power plants and lightweight hull materials was considered as a means of reducing power requirements for a given payload capacity.[1]

Reducing frictional resistance by sucking off the boundary layer of water through slots at intervals along the hull and pumping the water overboard has been proposed as a means of improving speed or reducing power requirements. However, it is felt that the gain in effective power due to the reduction of frictional resistance affected by the boundary layer control would not be large enough to offset the extra power required to provide the suction. Also, it is clear that substantial penalties in cost and complexity of internal hull layout and hull fabrication would result from use of such a boundary layer control scheme. With respect to the possibility of reducing frictional resistance by means of hull coating, it has been found that no significant drag reduction has been achieved in this manner in experiments to date. It is concluded that neither of these two methods holds much promise as a technique for significantly reducing frictional resistance of displacement hull ships in the time period of interest in this study.

A great deal of work has been done over the years to develop hull forms that minimize wave-making resistance and thereby reduce power requirements. There are limitations, however, on the practicality of certain hull forms. In addition to meeting the criterion of minimizing wave-making resistance, hull design must be compatible with the mission or the type of ship. Very fine lines, such as are typical of higher-speed vessels, are not well suited to cargo-type vessels. Clearly, inefficient layout or utilization of available space will characterize high-speed displacement hulls used as cargo vessels. Regardless, such penalties would have to be borne as part of the cost of displacement-type cargo vessels capable of very high speeds.

One of the possibilities for reduction of wave resistance through hull design is the use of a bulbous bow. Many seagoing vessels have bulbous bows of cross-sectional area equal to approximately 10 percent of midship section area. Such designs reduce horsepower requirements by about 8 percent. Very large bulbous bows have been suggested, equal to about 25 percent of midship cross section area. Significant reduction in wave-making resistance is achieved in this manner, but the reduction tends to be offset by the increase in frictional resistance due to increased wetted surface. The result is a net gain in available power of only about 7 percent (in the speed range of 18-20 knots).

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1. See Technical Notes at the end of the text.

In view of the possible disadvantages to be encountered in each particular approach and the fact that only marginal reductions in power requirements appear feasible with these several techniques for reducing drag, the characteristics and potential of high-speed displacement hull vessels have been projected on the assumption that high speed would be achieved basically through the application of very high power, using lightweight propulsion plants and a conventional hull form with fine lines. This appears to be the most realistic basis for projecting the feasibility of high-speed displacement hulls.

A hull form with a displacement-length ratio of 60 $\left(\frac{\Delta}{(.01L)^3} = 60\right)$ and a beam-draft ratio of 2.25 $\left(\frac{B}{H} = 2.25\right)$ has been used in estimating resistance and power requirements. This hull form has superior high-speed residual resistance characteristics and meets the requirements for strength and stability. Power requirements for displacement hulls with speeds up to 70 knots and total displacements up to 40,000 tons have been derived, indicating payload potentials for design ranges up to 3,000 miles.

The calculations indicate extremely high power requirements for large vessels of high design speeds relative to the maximum size of power plants in present naval vessels. It is evident that there is a limit to the amount of power for which effective propulsion systems can be designed, and that such a horsepower limit would establish a constraint on the maximum vessel size and speed that would be feasible in the time period of interest. (At this juncture it is assumed that cost is not a constraint.) The judgment was made that 600,000 horsepower probably was the maximum installed horsepower that should be considered feasible for the displacement hull vessel. The underlying assumptions are that it should be feasible to transmit and convert to thrust up to 100,000 to 150,000 horsepower per shaft, and that up to four to six shafts could be installed in larger vessels. Also, it is assumed that supercavitating propeller design can be further developed to the point at which such high-powered, high-speed propulsion systems are practical. These assumptions are perhaps optimistic, but they establish clear constraints on the maximum speed and payload potential of the advanced displacement hull.

The basic characteristic and performance data developed for the advanced displacement hull assume mild steel construction and comparatively lightweight, geared steam turbine propulsion. Improvements in the design potential of the displacement hull constructed of aluminum and propelled by gas turbine or nuclear propulsion are also examined. Detailed characteristics and performance charts are presented in Appendix A. Selected charts showing potentials for displacement hull vessels incorporating alternative hull materials and propulsion plants are included in this section.

Performance Potential

The maximum payload achievable at various operating speeds is shown in Fig. 3. The general constraint is a limit of 600,000 total installed shaft horsepower, as described above. The basic set of curves on the left indicates maximum payload potential as a function of speed at various ranges, assuming steel hull construction with geared steam turbine propulsion. For example, it is seen that the maximum feasible design speed for a 2,000-ton payload equivalent and a range of 3,000 miles is about 45.5 knots; at a range of 1,000 miles the maximum design speed is about 50 knots. The higher speed results from the fact that less fuel and fuel capacity are required (thus significantly reducing weight), and the power available (600,000 hp) can be used to achieve a greater speed in a somewhat smaller vessel (25,500 tons total displacement for 3,000-mile range and 45.5 knots versus 17,000 tons total displacement for a 1,000-mile range and 50 knots--compare Figs. A-25 and A-28, Appendix A).

The curve on the right of Fig. 3 indicates the maximum payload potential if the advanced displacement hull is constructed of aluminum and propelled by a lightweight marine gas turbine power plant. Again the basic constraint, which is controlling for payloads above 4,000 tons, is a maximum of 600,000 installed shaft horsepower. However, below a 4,000-ton design payload the maximum speed of aluminum hull vessels probably will be constrained to less than that theoretically achievable with this hull form and 600,000 horsepower. This is because there is danger of swamping smaller displacement vessels at very high speeds. A lower limit line on design payload is shown on the chart. This limit is actually a function of the length of a vessel and its speed. For the hull form used in this study, the speed-length ratio $\left(\frac{V}{\sqrt{L}}\right)$ cannot exceed 2.5 without danger of swamping. This ratio is not approached with the steel hull vessels because of the characteristically low payload-to-displacement ratios achievable at very high design speeds. To put it another way, a high-speed steel-hull vessel, even of extremely limited payload capacity, will necessarily be of sufficient size and length that a speed-length ratio of 2.5 is not approached. A steel hull vessel with a payload capacity of 500 tons and a 45-knot speed would have a total displacement of about 11,000 tons (2,000-mile range) and a length of 575 feet. By contrast, an aluminum hull vessel with the same speed, payload, and range would have a total displacement of under 2,000 tons and a length of 325 feet.[1]

1. Payload, speed, and displacement, assuming a 2,000-mile range, may be determined from Figs. 4 and 5. Hull dimensions as a function of total displacement may be read from Fig. A-37, Appendix A.

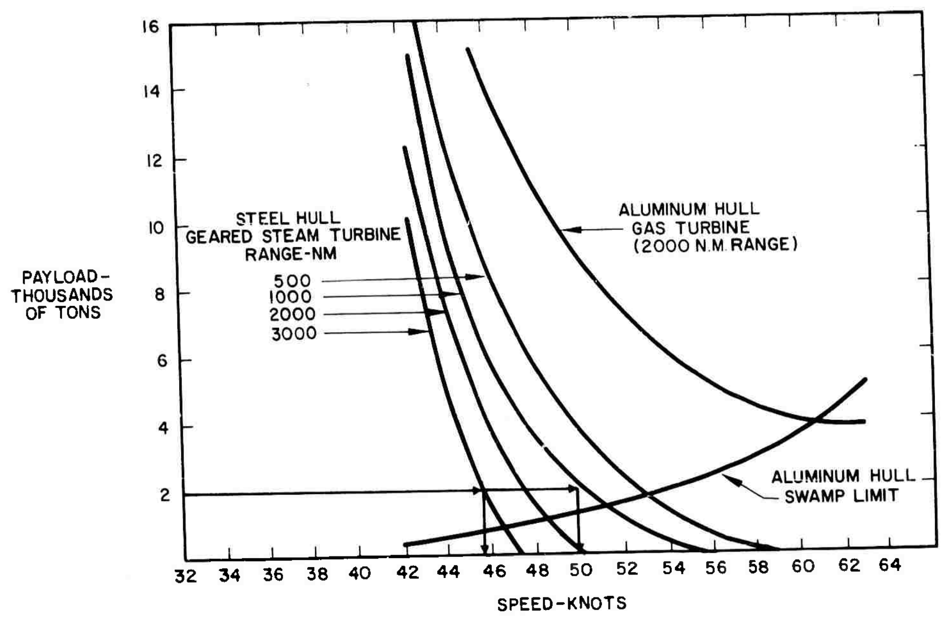

FIG. 3 ADVANCED DISPLACEMENT HULLS
MAXIMUM PAYLOAD VERSUS SPEED
(Constrained by a Maximum of 600,000 SHP)

A high percentage of total displacement is available for payload in an aluminum hull, lightweight gas turbine vessel in contrast to a steel hull, steam propulsion vessel. This will be seen in subsequent figures. (In Fig. 6 it will be seen that about 14 percent of the total displacement of a 10,000-ton steel hull vessel is available for payload at a 2,000-mile range; in Fig. 7 it will be seen that fully 50 percent of the total displacement of a 10,000-ton aluminum hull vessel is available for payload at the same range.) In Fig. 3 it is indicated that an aluminum displacement hull vessel with a payload capacity of 4,000 tons could have a design speed of up to 60 knots (assuming 2,000-mile range). Larger payload capacities (at the same range) would require a reduction in maximum design speed. Smaller payload vessels (under 4,000 tons) could be designed for the 60-knot speed only if excess capacity were built into them in order to have a vessel of sufficient size and length to avoid the danger of swamping. This is indicated clearly in Fig. 3.

In Appendix A is a series of charts which can be used in determining the total displacement and power of a displacement hull vessel of a specified speed and payload. Each chart is for a particular range and combination of hull materials and type of propulsion. These charts can be used to establisn the general characteristics of a particular-size vehicle and the possible range of payloads and speeds under a given set of assumptions. They can also be used comparatively to determine the influence of range or type of construction on the feasibility or characteristics of vessels of particular speed or size.

On Fig. 4, for example, are basic curves showing payload, displacement, speed, and horsepower for steel hull, geared steam turbine displacement hulls with a design range of 2,000 miles. The effect of increasing design speed on the power requirement and total displacement is well illustrated. At 35 knots a ship with a 2,000-ton payload capacity would require 110,000 to 120,000 shaft horsepower and would have a total displacement of 8,000 tons. A 10-knot increase in design speed to 45 knots would increase shaft horsepower to almost 425,000 and total displacement to over 17,000 tons, assuming the same 2,000-ton payload equivalent. A 5-knot increase from 45 to 50 knots would further increase power plant requirements to about 800,000 horsepower (well over the adjudged limit of 600,000) and total displacement to almost 27,000 tons. As can be seen, speed is costly in terms of power and vessel size. In a later section these costs are assessed in dollar terms.

The effect on speed and payload potentials of using aluminum construction and lightweight gas turbine propulsion is seen in comparing Fig. 5 with Fig. 4. For example, again assuming a 2,000-ton payload, it is shown that total power requirements at 45 knots for an aluminum hull

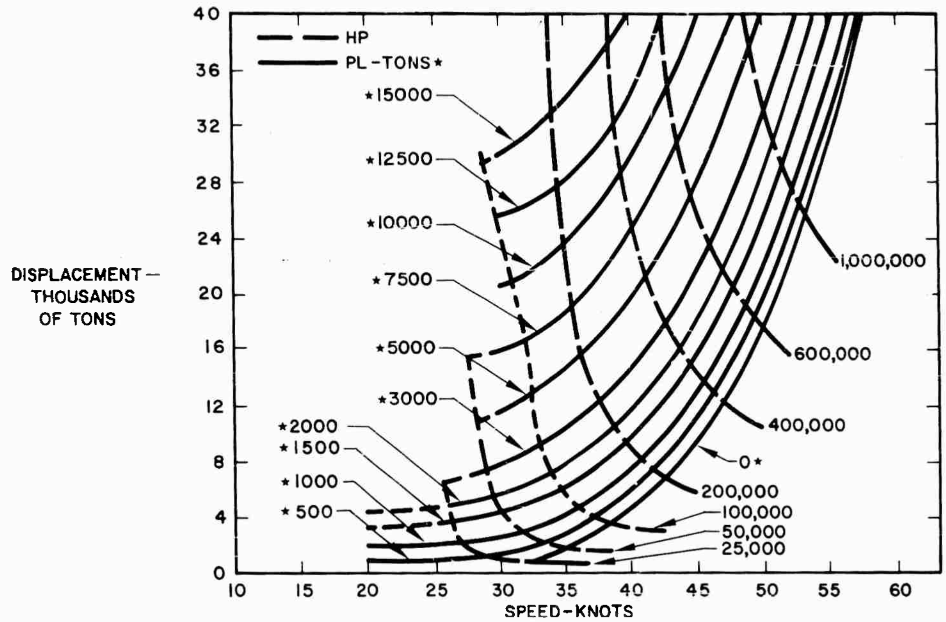

FIG. 4 ADVANCED DISPLACEMENT HULLS, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER, MILD STEEL HULL GEARED STEAM TURBINE POWER PLANT

(Range, 2,000 Nautical Miles)

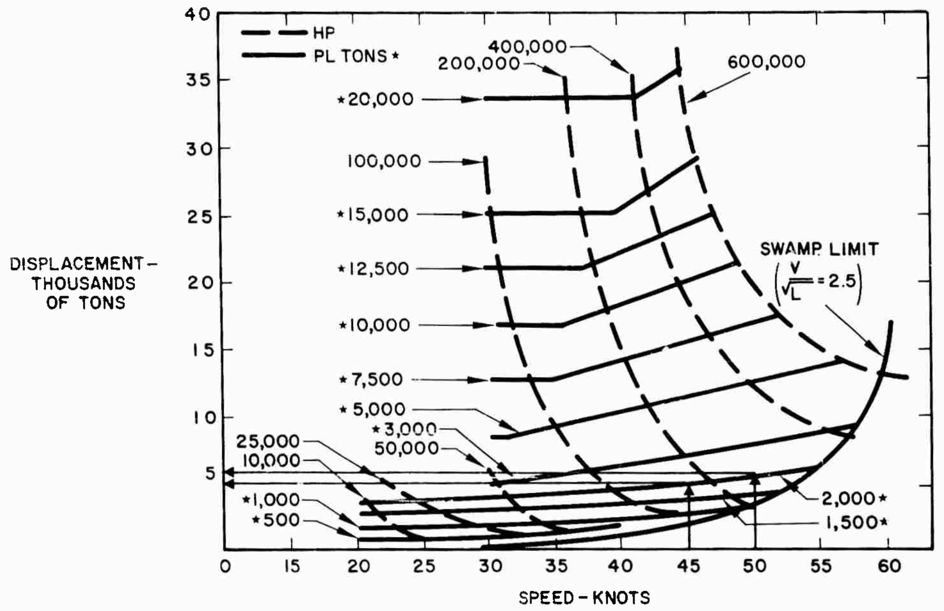

FIG. 5 ADVANCED DISPLACEMENT HULLS, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER, ALUMINUM HULL GAS TURBINE POWER PLANT (Range, 2,000 Nautical Miles)

vessel would be well under 200,000 shaft horsepower (compared with about 425,000 for the steel hull vessel), and total displacement would be only 5,000 tons (compared with 17,000 tons for the steel hull). A 5-knot increase in design speed to 50 knots would increase the power requirement only to about 250,000 horsepower and the total displacement only to well under 6,000 tons.

The very significant differences in the size of comparable steel and aluminum hulls, powered respectively by steam turbine and gas turbine power plants, relate to both hull weight and power plant weight. The geared steam turbine power plant assumes a unit propulsion plant weight of 20 pounds per shaft horsepower (the projected minimum achievable weight for such systems--see page A-6, Appendix A). The gas turbine propulsion system assumes a unit weight of 5 pounds per shaft horsepower. The weight of an aluminum hull is about one-half that of a steel hull. In addition, in deriving the characteristics and weight breakdown of the aluminum hull vessels, it was felt that greater concern would be given to weight conservation in aluminum hull design than in steel hull design. Consequently, it was considered that a service margin of only 15 percent in installed power (instead of 30 percent as for the steel hull vessel) would be appropriate and that an allowance of only 10 percent of total displacement for outfit weight would be reasonable for aluminum hull vessels. Regardless of whether steel or aluminum design is used, it is believed that outfitting weights and weights of auxiliaries can be significantly reduced from present levels. Greater weight-consciousness in design will be of importance in developing high-speed vessels.

Further appreciation of the differences in payload potential between the steel hull and aluminum hull displacement vessels may be gained in comparing the weight breakdowns for the two alternatives. Figure 6 shows payload potential and weight breakdown as a percentage of total displacement for 40-knot displacement hulls up to 40,000 tons displacement. As shown, ranges vary from zero range (no fuel) to a range of 3,000 miles. The payload potential is that portion of total displacement not allocated to weight of hull, propulsion, auxiliaries, outfitting, or fuel. Fuel requirements vary with range and, as a result, design payload does likewise. For example, as shown in the figure, a 10,000-ton vessel with a design range of 2,000 miles would have a payload potential of about 14 percent, or 1,400 tons. At smaller displacements it can be seen that range or payload or both may be severely limited.

The weight breakdown of a 40-knot aluminum hull vessel is shown in Fig. 7. Again, payload potential at various ranges is shown as a percentage of total displacement, and it may be seen that a 10,000-ton aluminum vessel has a payload potential of about 50 percent, or 5,000 tons, as

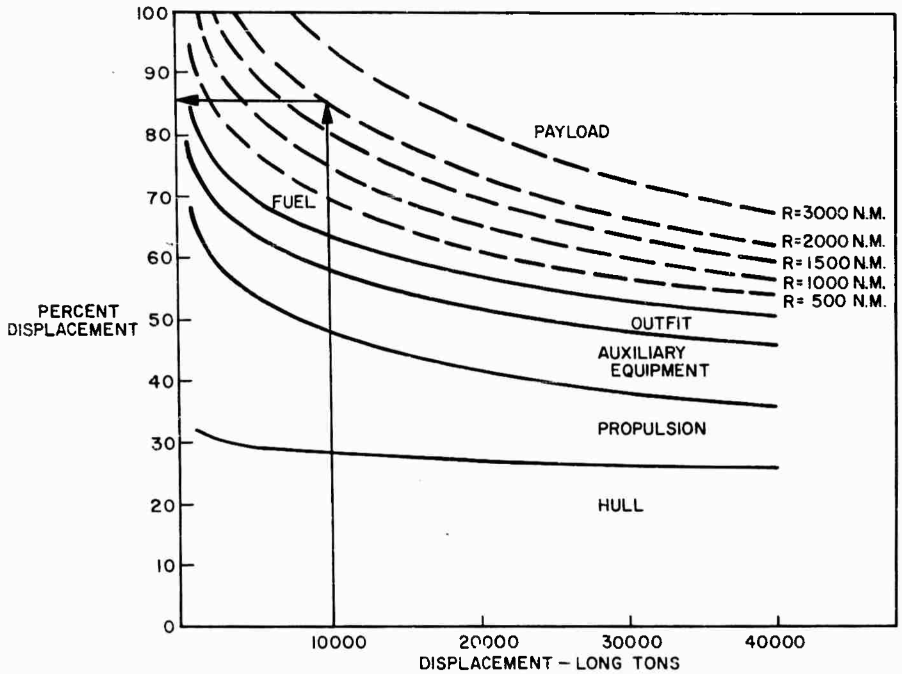

FIG. 6 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, MILD STEEL HULL GEARED STEAM TURBINE POWER PLANT
(Speed, 40 Knots; Various Ranges)

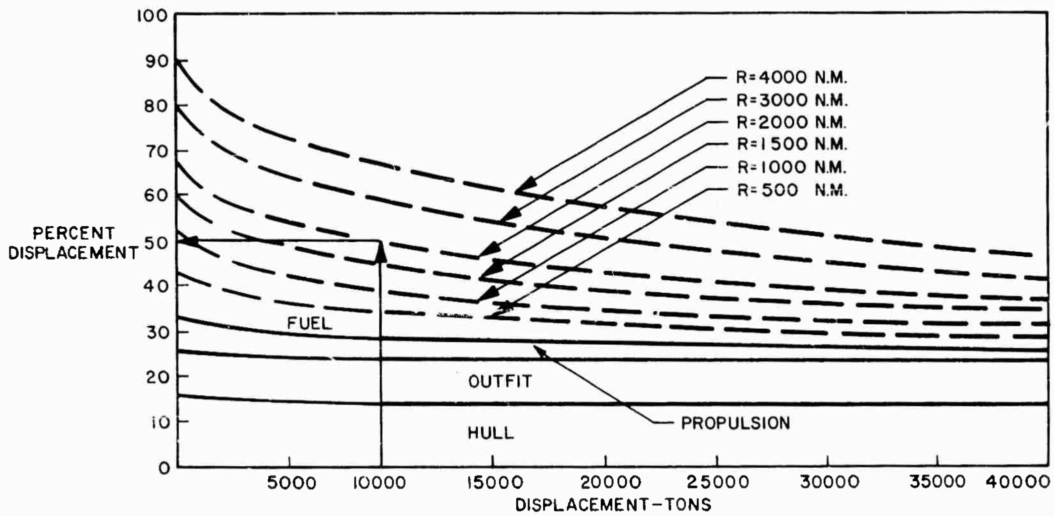

FIG. 7 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT ALUMINUM HULL, GAS TURBINE POWER PLANT (Speed, 40 Knots; Various Ranges)

compared with 1,400 tons for the steel hull vessel. It is evident that the feasible scope of range and payload characteristics for high-speed vessels is much less restrictive with aluminum hull design than with steel hull design.

Combining the aluminum hull with the lightweight gas turbine is a most spectacular way to save weight. As yet, no large ships have been fabricated fully of aluminum, although several smaller all-aluminum craft are in use by the Navy, and many ships have extensive aluminum components. Aluminum is superior to steel in corrosion resistance, but aluminum structural members, having only one-third the modulus of elasticity, permit approximately three times more deflection than steel members of equivalent strength. Yield stresses of aluminum compare favorably with those of mild steel, and future development may result in higher-strength alloys. Material costs of aluminum run six to eight times those of mild steel (total vessel cost comparisons are given later). Some use of high-strength steel in hull frames, deck plates, foundations, and the like could result in measurable weight savings in steel hull vessels; however, achieving appreciable weight savings requires the use of a lightweight material of strength equivalent to that of mild steel, such as aluminum.[1]

Nuclear propulsion has been investigated for both steel and aluminum hull displacement vessels. The unit weight of existing nuclear propulsion plants is high, running from 90 to 100 pounds per shaft horsepower. Present marine-type nuclear reactors are of two types, the boiling water reactor and the pressurized water reactor, both using steam turbine propulsion machinery. The organic moderated reactor is still in the development stage, but its specific weight appears to be about the same as that of existing marine reactors. It is felt that lightweight, gas-cooled reactors, based on developments arising out of the aircraft nuclear power program, show great promise, with possible weight-power ratios of from 7 to 45 pounds per shaft horsepower, depending on size. Specific weight decreases significantly as horsepower rating increases and as gas turbines are substituted for steam turbines. In the derivation of weight breakdowns for nuclear powered vessels, it has been assumed that nuclear powerplant weight per horsepower would be as shown in Fig. A-38, Appendix A. This chart shows the specific weight of nuclear-powered propulsion systems for marine use decreasing to about 20 pounds per horsepower for plants rated at 100,000 horsepower or more.

In Fig. 8 the payload potential and weight breakdown of steel hull displacement vessels powered by lightweight nuclear power plants is shown

1. See Technical Notes at the end of the text.

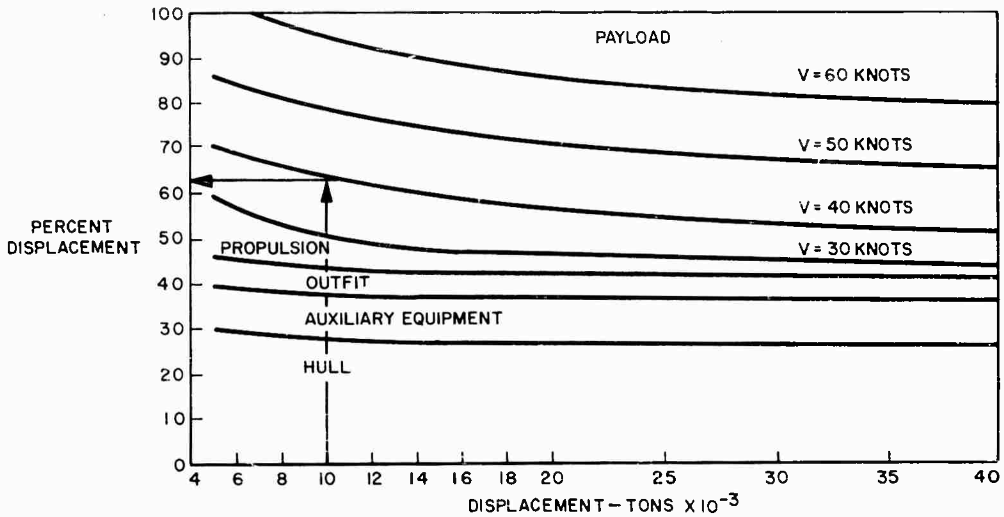

FIG. 8 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, MILD STEEL HULL NUCLEAR POWER PLANT

(5,000 to 40,000 Tons Displacement; Various Speeds)

for speeds up to 60 knots. In the example sketched on the graph it may be noted that the payload potential of a 10,000-ton displacement vessel with a 40-knot design speed would be about 36 percent of displaeement, or 3,600 tons. This compares to a 1,400-ton payload potential for a steel hull, steam turbine vessel of the same displacement and speed (Fig. 6) and a 5,000-ton payload potential for an aluminum hull, gas-turbine-propelled vessel of the same total displacement and speed (Fig. 7). In both of these cases the design range is 2,000 miles. The nuclear-powered vessel would have essentially unlimited range. Again, there are signifieant differences in costs, as will be seen later. It is of significance to note that for smaller ships the payload potential of nuclear-powered vessels, as a pereentage of total displacement, decreases sharply due to high shielding weight. Shielding weight per unit of power decreases substantially with larger power plants. This may be seen in Fig. A-39, Appendix A, which shows the weight breakdown for vessels up to 5,000 tons displacement (Fig. 8 shows only displacements from 5,000 to 40,000 tons). It may be observed in Fig. 8 that at a 60-knot speed the payload potential of the steel hull nuclear propelled vessel would be very limited--only 500 tons for a 10,000-ton displacement vessel. This potential inereases to about 2,600 tons at 20,000 tons displacement and to 8,000 tons for a 40,000-ton vessel. However, the power required for a 60-knot design speed in a 40,000-ton vessel is close to 1.75 million horsepower, or almost three times the maximum of 600,000 horsepower judged to be feasible at the present time. The limited design payload potential at very high speeds, even with relatively large displaeements, is the result of the extraordinary amount of power required to achieve significantly higher speeds in a displacement hull design and the heavy weight of hull and machinery required to provide such power.

In view of the above, a possibility that should be considered is the potential of a nuclear-powered aluminum hull vessel. Figure 9 shows the payload potential and weight breakdown projected for such vessels. It is observed that a 10,000-ton vessel with a design speed of 40 knots would have a payload potential of about 58 percent, or 5,800 tons; this compares with a payload potential of 3,600 tons for a steel hull nuclear-propelled vessel, a potential of 5,000 tons for the gas-turbine-powered aluminum hull, and a potential of 1,400 tons for the steam-turbine-powered steel hull. These alternatives are compared in Section VIII in terms of total power requirements and construction costs for various-size platforms.

The calculations and data presented on the aluminum hull vessels indicate that the weight of a light ship could be as low as 25 to 35 percent of displacement. Because of stability considerations, it is assumed that at no time should such vessels be allowed to becomc lighter than 40 percent of total displacement. Payload and fuel, plus salt-water

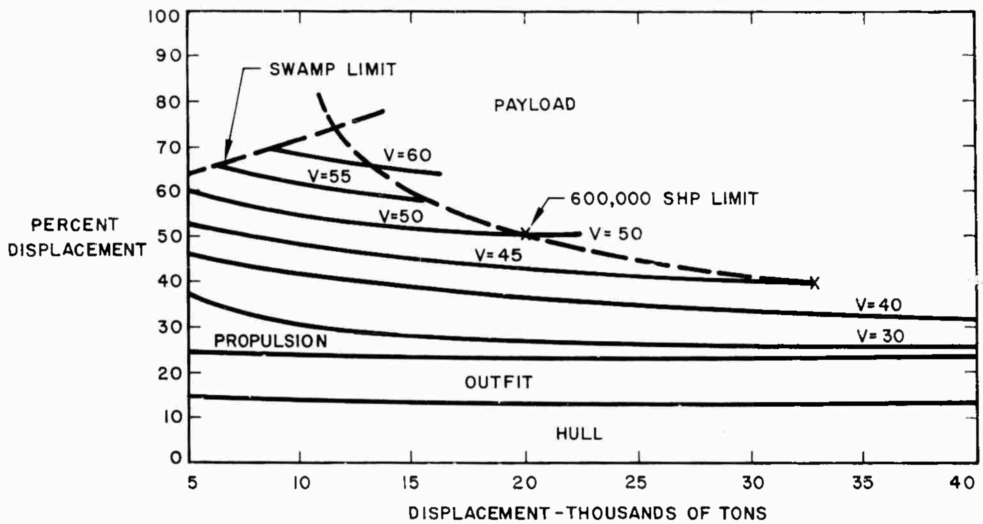

FIG. 9 ADVANCED DISPLACEMENT HULLS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, ALUMINUM HULL, NUCLEAR POWER PLANT
(5,000 to 40,000 Tons Displacement; Various Speeds)

ballasting when necessary, could meet such an operational requirement. It is indicated strongly, however, that further detailed study of the stability characteristics of the hull form described here, particularly in reference to the design of high-speed, lightweight hulls, is warranted if the potential performance of the advanced displacement is found interesting. It is also concluded that preliminary hull layouts should be undertaken for advanced displacement hulls of a payload and speed potential of particular interest. Because of fine hull lines, available volume and deck space may limit practical utilization of payload potentials derived on a weight basis (especially in the case of lightweight hulls). A number of possible layouts should also be undertaken to establish the most feasible and effective machinery and propulsion plant arrangements for vessels designed for very high speeds.

IV PLANING HULLS

Utilization of planing hull designs is a possible means of reducing the hull resistance of high-speed ships relative to resistance encountered by vessels with large draft displacement hulls. Reductions in hull resistance achieved with a planing hull design would reduce power requirements for high-speed propulsion relative to the requirements associated with a displacement hull of equivalent size and speed. Thus far, however, the planing hull, an example of which is the PT boat of World War II fame, has been limited in size to well under 100 tons total displacement. Very little design or model hull testing has been done for large (250-ton payload equivalent or larger) planing hulls. Projections of the possible characteristics of large planing hulls have been based on extrapolation of existing design information pertaining to smaller boats taking into account probable limitations in the laws of similarity in relating the results of model testing to full-size hull performance.

Hull Form and Power Requirements

There appear to be two major limitations on the potential size and speed of planing hull vessels: (1) the lowest speed at which the planing hull becomes advantageous in reduced resistance, compared to the speed of displacement hull forms such as those of destroyers, is quite high for large vessels and increases as total displacement increases, and (2) the speed of planing hulls, by virtue of mode of operation, is highly affected by rough water conditions.

With regard to the first point, it is noted that large planing hull vessels will draw substantial draft, and at high speed, while favorable compared to the speed of displacement hulls, the absolute power requirement remains high. Again, as in the case of displacement hulls, there are limits on the maximum power convertible to thrust and limits on hull space available for large-capacity propulsion systems. The judgment has been made that the maximum power feasible for installation in the planing hull is 400,000 horsepower. This assumes that up to 100,000 horsepower per shaft could effectively be converted to thrust. However, it is considered that, in contrast to the displacement hull which might employ six shafts, even large planing hulls should be limited to a maximum of four shafts. Because of the criticality of hull lines in the planing hull, if indeed it is to operate as a planing hull, it is believed that there would be less flexibility in possible propeller and machinery layout and that four shafts would probably be a maximum.

On the second point, which relates to the sensitivity of planing hull design requirements (and operational performance) to the sea state, it is considered that planing hulls of larger than, say, 3,000 tons total displacement would be impractical. This may be an optimistic judgment, but it is evident that there is an upper limit on the size of planing hull designs. Extreme structural problems may be encountered in designing very large vessels to withstand the stress of slamming. At high speed there will be a considerable tendency for the planing hull to slam, creating high local stresses and perhaps unacceptable crew discomfort even when operating under moderate sea conditions (state 3 seas, for example). One of the major factors limiting the potential of high-speed planing hull designs for major fleet units is the evidence that high-speed operations would be feasible only under the most favorable sea conditions. In high-speed operation, alternative platform concepts are less sensitive to adverse sea conditions than the planing hull, as will be shown.

In the projection of probable performance characteristics, it has been assumed that aluminum hull structure would be employed along with very lightweight gas turbine propulsion. The speed potential of the planing hull is substantially higher than that of displacement hulls but the feasibility of achieving high speeds within allowable limits of power and total displacement depends on a high degree of weight-consciousness in design. It has been assumed that specific weight of gas turbine propulsion systems would be on the order of 3 pounds per horsepower (gas turbines with weights of from 1 to 10 pounds per horsepower are available). Nuclear propulsion was also considered. The basis for developing the performance characteristics shown in Figs. 10 through 13 below are discussed in detail in Appendix B.

Perfor ce Potential

The maximum payload and speed potentials that appear potentially achievable with the planing hull design concept for high-speed platforms are illustrated in Fig. 10. Maximum payload potential is plotted as a function of speed, and three possible constraints are indicated. First is the constraint that the total displacement of a planing hull might be limited to a maximum of 3,000 tons. This appears to be optimistic but, even taking this optimistic view, it is seen that payload potentials are limited to the equivalent of those of destroyer-type vessels. It is also seen that payload potential falls sharply with increase in design speed. Note that a design range of only 500 miles is specified in the figure. At greater range, the payload potential is sharply reduced, depending on the particular range.

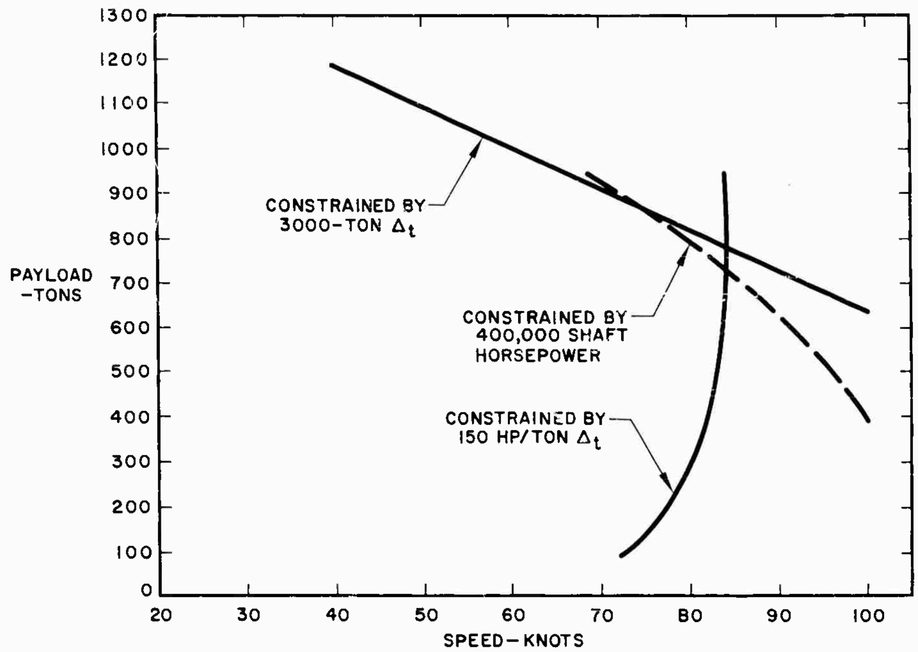

FIG. 10 PLANING HULLS
MAXIMUM PAYLOAD VERSUS SPEED
(Range, 500 Nautical Miles)

A second constraint on the possible size and speed of the planing hull is maximum allowable power. A limit of 400,000 horsepower is shown in Fig. 10. This limit is not reached below a speed of 70 knots for payload potentials up to 900 tons. Above the 900-ton payload equivalent, the constraint on further increases in design payload at the 70-knot speed is imposed by the 3,000-ton limit on total displacement.

At about 80 to 85 knots, it is seen that a third possible constraint on further increases in design speed is encountered. This is a limit of 150 horsepower per ton of total displacement. This is a fairly dense power loading, but it should be achievable with lightweight propulsion systems.

The characteristic relationships between speed, power, payload, and total displacement of planing hull vessels which establish the speed and payload limits shown above are indicated in Fig. 11. This particular chart is for a range of 1,000 miles. Similar charts for other ranges are provided in Appendix B. Again, the chart illustrates the comparatively limited payload potentials of the planing hull. Absolute limits on speed potential are reached between 85 and 100 knots at displacements ranging from 500 to 3,000 tons.

The extreme sensitivity of the planing hull speed and payload potential to design range is illustrated in Fig. 12. This figure shows payload potential and major weight components as percentages of total displacement, assuming in this particular chart a design speed of 50 knots and gas turbine propulsion. Comparable charts for other design speeds are given in Appendix B. Range is extremely limited, even at this comparatively slow design speed. For example, at a 3,000-ton displacement the maximum payload potential for a 1,500-mile range (about the range of a destroyer) would be only a little over 300 tons. To achieve the minimum payload equivalent of a destroyer (750 tons) with a design speed of 50 knots, range would have to be cut to under 1,000 miles.

In view of the limit on payload and range potential, the feasibility of nuclear power was investigated. The results are indicated in Fig. 13, which shows payload potential and major weight components for nuclear-powered planing hulls of speeds of 40 to 80 knots. For smaller vessels, a very high proportion of power plant weight is accounted for by weight of shielding. For larger vessels and higher horsepower plants, the unit weight of shielding (and total power plant) is greatly reduced. Payload potential for smaller vessels is not significant because of high power plant weight, as shown. At a 3,000-ton displacement and a 50-knot speed, the payload capacity is almost 550 tons, but at higher design speeds it decreases sharply. For the reasons indicated above, nuclear power becomes

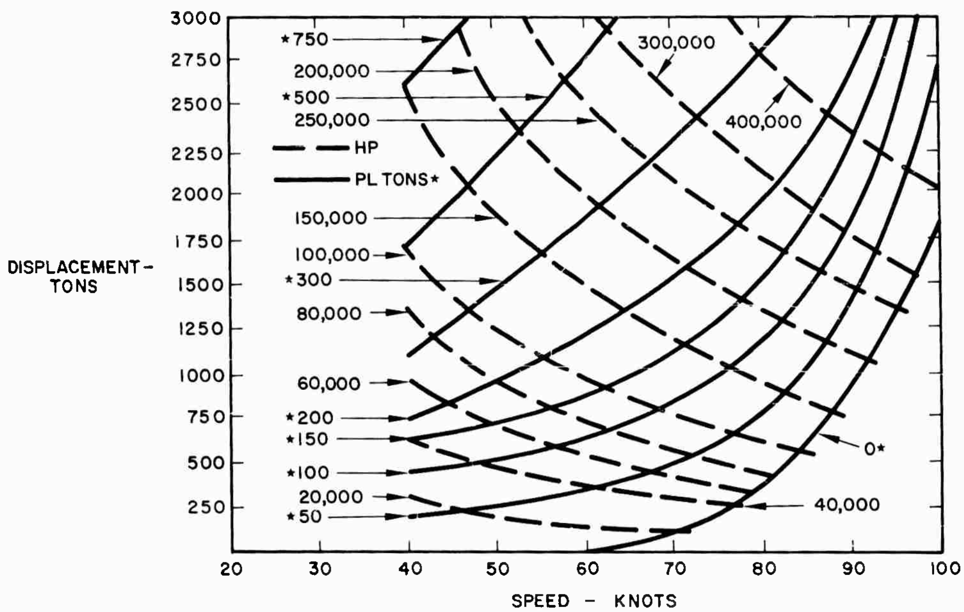

FIG. 11 PLANING HULLS, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER, GAS TURBINE POWER PLANT
(Range, 1,000 Nautical Miles)

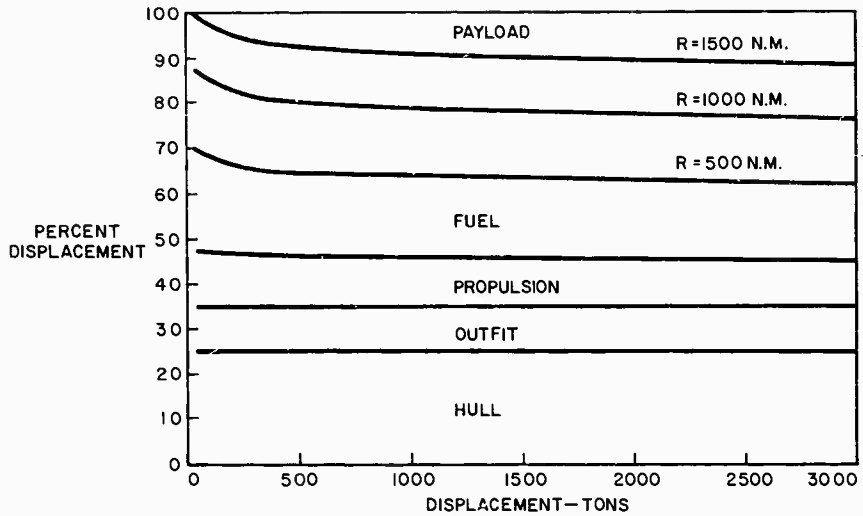

FIG. 12 PLANING HULLS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Speed, 50 Knots; Various Ranges)

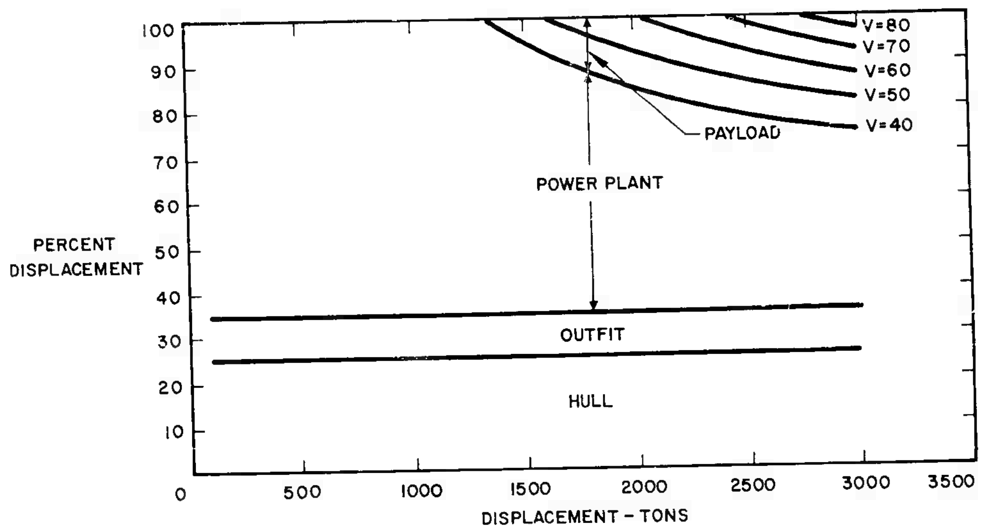

FIG. 13 PLANING HULLS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, NUCLEAR POWER PLANT
(Speed, 40 to 80 Knots)

more attractive (in terms of allowable payload potentials) for larger vessels, as would be indicated by further extrapolation of the curves shown in the figure. As concluded, however, larger planing hulls do not appear feasible from a structural and operational standpoint. The potential attractiveness of planing hull speeds is offset by the limitations on their use under rough water conditions, as indicated.

V HYDROFOILS

A careful review has been made of various studies analyzing the feasibility of ocean-going hydrofoil vessels. Certain of these studies appear to be unduly optimistic with respect to prediction of performance potential, particularly as it relates to achievable lift-to-drag ratios and weight of hull structures (see Appendix C). Nevertheless, the hydrofoil concept is a potentially attractive means of achieving high-speed capabilities in ocean-going vessels. When foilborne the hull of the hydrofoil vessel is free of the water, thus avoiding the enormous resistances encountered at high speed by displacement hulls and, to a lesser extent, planing hulls. As a consequence, power requirements would be much smaller for hydrofoil vessels than for displacement hulls and planing hulls of comparable payload and speed.

Very active and significant development work is under way in the hydrofoil field, and a number of prototype craft are in operation or sea trials. However, designs to date have predominantly been for small vessels relative to the size and payload potentials of interest in this study. Therefore, a prime objective has been to assess the feasibility of hydrofoil vessels of a size or payload potential that would be characteristic of major units in amphibious task force operations in the 1975-1980 period.

Foil Systems and Power Requirements

The hydrofoil concept offers the potential for speed ranges from 25-30 knots up to 100 knots or more. For lower speeds, up to 50 to 70 knots, subcavitating foil systems are employed. At higher speeds, however, supercavitating foils would be required. Power requirements are dependent on the ratio of lift to drag (L/D) achievable in any given design and foil system. The L/D ratio achieved is a function of the speed and size of the vessel. There is, however, a discontinuity in the transitional area (V = 50-70 knots) between subcavitating and supercavitating systems. This will be noted in the performance and power charts described later in the section.

In this study, power requirements have been derived assuming L/D = 528/V. This is some improvement over the ratio recommended by at least one authority for good practical designs (see Appendix C) but is much more conservative than that recommended or used by others. An L/D of 528/V is believed to be reasonable as a basis for projection of the performance characteristics of hydrofoils that might be designed and

constructed in the time period of interest here. No effort has been made in the parametric curves showing possible speed and payload characteristics of hydrofoils to differentiate L/D for different sizes of vessels. As a result, the estimates of performance achievable with larger craft may be somewhat conservative, since the achievable lift-drag ratio tends to improve with the size of vessel.

Horsepower requirements have been derived initially for foilborne cruise condition. For takeoff, the hydrofoil will require from 25 to 50 percent more power, depending on sea conditions. The time required for takeoff is less than 60 seconds; therefore, the engines and propulsion system should be capable of developing a maximum shaft horsepower of 1.5 times the required continuous horsepower rating for a duration of at least 60 seconds.

Basic curves have been developed for hydrofoils powered by lightweight aircraft-type gas turbine power plants and also by nuclear power plants. Water propellers have been selected as the most feasible thrust device within the current and near-term state-of-the art. Power requirements for hydrofoils of an over-all size judged feasible (discussed below) are quite low by comparison with those for planing hull and displacement vessels. However, in the case of the hydrofoil, water-propeller propulsion systems require angle-jointed shafting. The maximum power per shaft on such systems might well be no greater than 50,000 horsepower, and this is probably a very high limit in comparison with the maximum power presently transmitted by angle-jointed shafting systems. A requirement for larger hydrofoil vessels would be the development and testing of high-power transmission and shafting systems or the development of alternate thrust devices, such as the water jet.[1]

The major limitation or constraint on the design payload potential of the hydrofoil would appear to be the maximum size of hull feasible of construction from the standpoint of the supporting foil and strut system. The weight of foil and strut systems increases disproportionately to increases in total gross weight of a hydrofoil. This will be illustrated later. Also, there may be significant weight penalties (and cost penalties) in designing the hull structure of larger vessels to permit support of the entire weight of the vessel on the struts when in foilborne operation. A detailed discussion of basic assumptions and methods used in calculating weight breakdown and performance potential for hydrofoils of various speeds and payloads is given in Appendix C.

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1. See technical notes at end of the text.

Performance Potential

The maximum payload and speed that it appears feasible to consider for hydrofoil designs in the time frame of interest here are indicated in Fig. 14. Payload potential as a function of speed is shown for subcavitating hydrofoil vessels of speeds up to 60 knots and ranges of 500, 1,000, and 1,500 miles. Shown also is the feasible limit on payload and speed for nuclear powered, subcavitating foil, hydrofoil vessels. The basic constraint is the assumption that the maximum-size hydrofoil would be about 3,000-ton total displacement, or gross weight.

The speed at which supercavitating hydrofoils should be used would be between 50 and 60 knots; however, for this study the discontinuity between the subcavitating and supercavitating designs is shown at an upper limit of 60 knots. The payload potential for the subcavitating hydrofoil is quite limited, even for lower design speeds. At the higher speeds (with supercavitating foils), it is seen that payload potentials are extremely limited, even for short ranges.

In Fig. 15 are shown the basic relationships among speed, horsepower, payload, and total displacement for both subcavitating and supercavitating hydrofoils. It is observed that in the higher speed ranges (60-90 knots) for a given payload potential there is a maximum speed achievable, regardless of further increases in installed power. Beyond the critical speed, further increases in power are so costly in terms of weight that speed capacity is in fact reduced. For example, the speed of hydrofoils with a 60-ton design payload can be increased up to about 81 knots by increasing power; further increases in power actually decrease speed capability.

In the case of the slower-speed hydrofoils (subcavitating foils) it appears possible to increase the design speed of vessels of any given payload by further increases in installed power but, as the curves indicate, increasingly smaller increments of speed are achieved and only at increasingly higher costs in installed horsepower. The power required per unit of increase in design speed increases sharply for hydrofoils of higher payload capacity. Note that Fig. 15 specifies a range of 1,000 miles. At higher ranges, potential payloads and speeds are even more restrictive. Data on other ranges are presented in Appendix C.

The sensitivity of payload potential to design range is illustrated in Fig. 16, which shows payload potential for a 50-knot hydrofoil as a percentage of total gross weight, or displacement. The chart graphically illustrates that payload potential is severely limited, even for larger hydrofoils, because of the rapid increase in percentage of total weight accounted for by foils and struts. Similar charts showing the weight

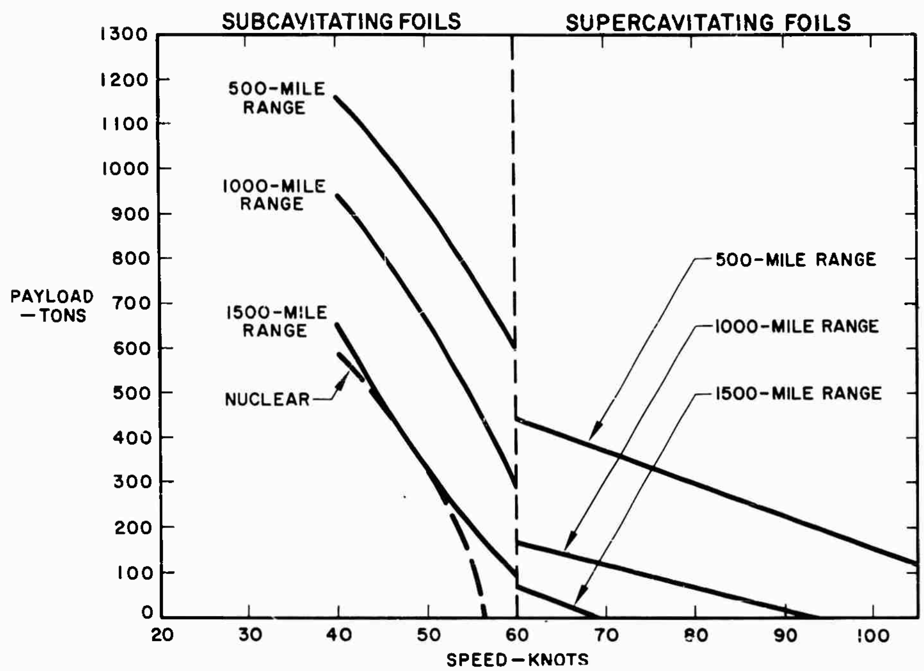


FIG. 14 HYDROFOILS
MAXIMUM PAYLOAD VERSUS SPEED
(Constrained by a Limit of 3,000 Tons
Total Displacement ($\triangle_t$))

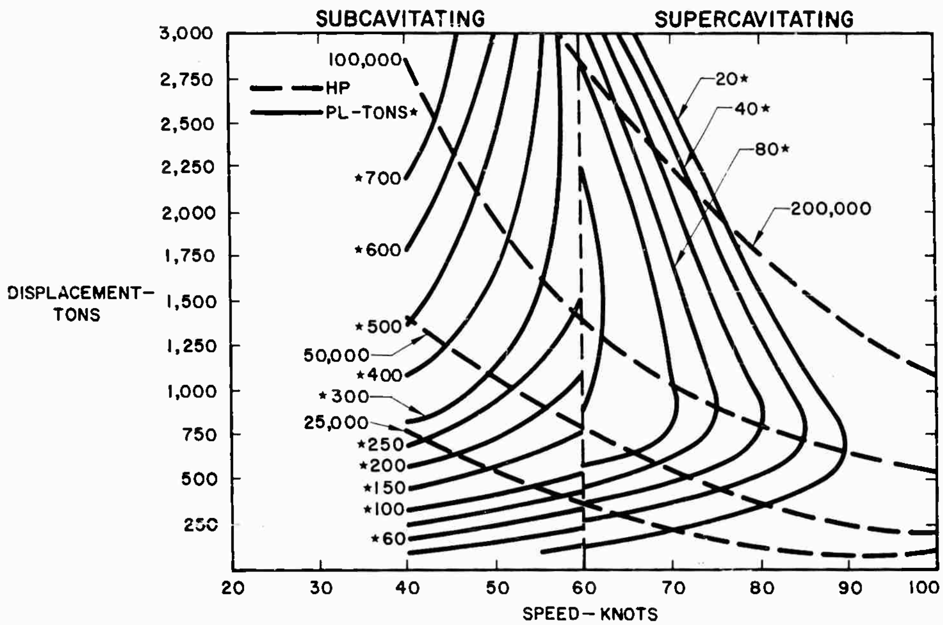


FIG. 15 HYDROFOILS, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED HORSEPOWER
(Range, 1,000 Nautical Miles)

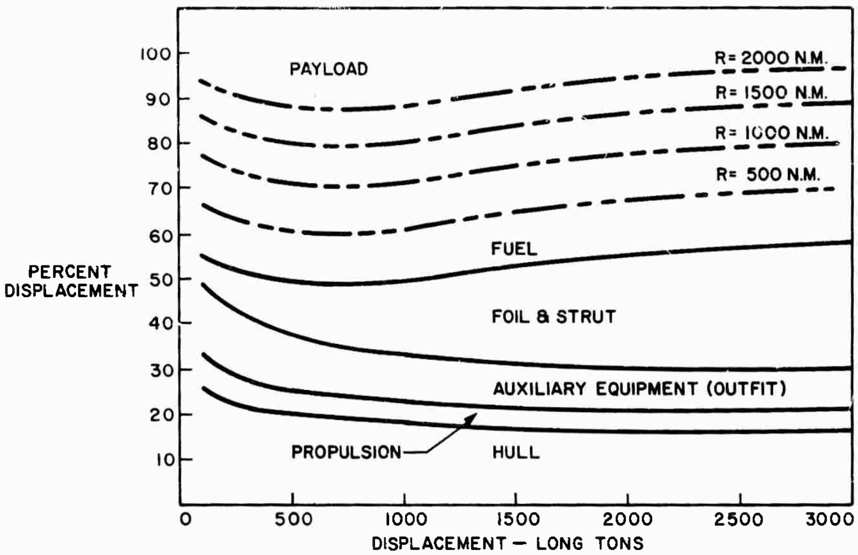


FIG. 16 HYDROFOILS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 50 Knots; Subcavitating Foils)

breakdown for hydrofoils of other design speeds are given in Appendix C. At higher speeds, it is found that foil and strut weight is an even greater percentage of total displacement, and both range and payload potential are curtailed accordingly. This is indicated in Fig. 17. This figure shows the payload and range potential of 80-knot hydrofoil designs of displacements up to 3,000 tons. The payload potential of a 3,000-ton hydrofoil is seen to be only 9 percent of total displacement, or 270 tons, at a design range of only 500 miles. At a 1,000-mile range, the maximum size of a hydrofoil would be about 1,800 tons, and payload capacity would be zero.

Severe range limitations in the hydrofoil powered by lightweight gas turbine propulsion systems suggest that lightweight nuclear propulsion might be attractive. However, as in the case of planing hulls, the hydrofoil concept appears to be suitable for only comparatively small vessels. In small vessels the high relative weight of the nuclear plant per unit of power limits the payload potential achievable. This is illustrated in Fig. 18, which shows the weight breakdown for a 50-knot nuclear powered hydrofoil. This figure, which may be compared directly with Fig. 16 (50-knot gas turbine hydrofoil), indicates that nuclear propulsion would be infeasible for vessels smaller than 1,800 tons and that, at that displacement, there would be no payload potential. For a 3,000-ton vessel, the payload potential would be only 300 tons, as shown. For faster vessels, the payload potential of nuclear powered hydrofoils would be even less, unless larger vessels could be constructed.

Closed-cycle, gas-cooled nuclear reactors have been considered for installation both above water and under water. Figure 18 was based on use of an above-water installation (see Appendix C, pages C-10 and C-24-25). An under-water nuclear plant mounted in a pod would be of significantly less specific weight than the above-water installation because of reduced shielding requirements, but this relative advantage would probably be lost because of increased total water drag from the pod and strut as compared with the alternative installation. The under-water pod for nuclear power has not been considered feasible as a design concept for the near-term period.

The costs of hydrofoil craft over the range of feasible design speeds are compared with the costs and characteristics of alternative platforms of the same payload capacities in Section VIII.

The hydrofoil is like the planing hull in that it appears to have a potential for only smaller payload missions, unless much larger-size hydrofoils become feasible than are now expected. Unlike the planing hull, the hydrofoil has good capabilities to operate under adverse sea

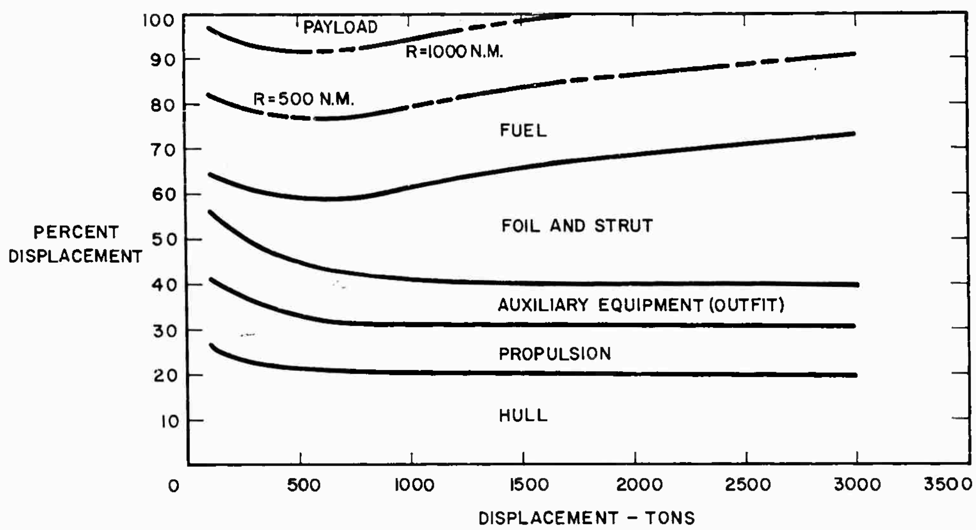


FIG. 17 HYDROFOILS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT (Speed, 80 Knots; Supercavitating Foils)

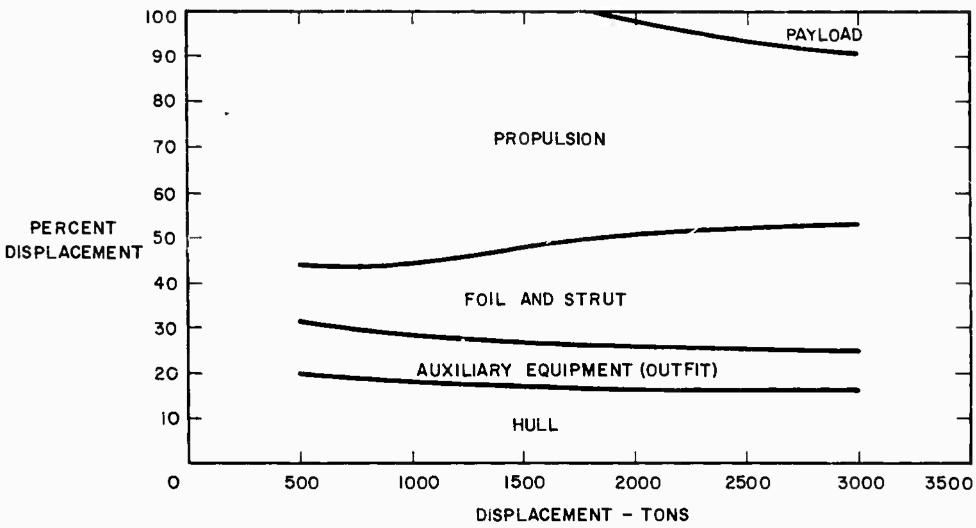


FIG. 18 HYDROFOILS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT NUCLEAR POWER PLANT (Speed, 50 Knots; Subcavitating Foils)

conditions. The maximum sea state in which the hydrofoil can operate is dependent primarily on strut height. In the performance charts developed in this study, it was assumed that hydrofoil design should permit the vessel to operate in sea states of 5 to 6.

VI GROUND EFFECT MACHINES

The ground effect machine (GEM) is a relatively new transportation concept potentially capable of offering significant advantages as a high-speed, large-capacity, ocean-going platform. The GEM, which always operates in close proximity to the surface of the ground or water, is supported on a cushion of air maintained by high-volume fans. Conceptually, the GEM employs the ground effect phenomenon (the augmentation in lift obtained by an airborne vehicle when operated in close proximity to the surface) to reduce the amount of power required relative to that which would be required to support the vehicle out of ground effect.

At speeds above 60 knots or so, the GEM would generally appear to require less power than alternative platforms of comparable payload potential (depending on the operating height of the GEM). Also, the efficiency of the GEM tends to improve significantly at larger design payloads. The GEM would appear to be the only concept that affords at least a potential for the design of platforms with payload equivalents comparable to those of major units in the present amphibious task force and with speeds of up to 100 knots or more.

Because the GEM operates free of the surface, supported on a cushion of air, it is inherently amphibious. Accordingly, the GEM could offer significant advantages for particular types of missions in the amphibious task force. Employment of GEM LST's or AKA's, for example, could have a great impact on the present concept of ship-to-shore operations. Moreover, since the GEM has very high speed capabilities, it could have a very significant impact on the entire concept of amphibious operations, including patterns of deployment, types of task force organizations, means of projecting landing forces ashore, and defensive requirements both en route and in an objective area.

A great deal of theoretical work has been done on the possible performance of GEM's, and numerous small payload GEM's of both experimental and operational designs have been built and successfully operated. However, very little work has been done to date on design and model testing of large ocean-going GEM's having the payload potentials of primary interest in this study. Nevertheless, on the basis of theoretical works and existing design and performance data for smaller GEM's, estimates of the probable power and weight characteristics of larger GEM's have been derived.

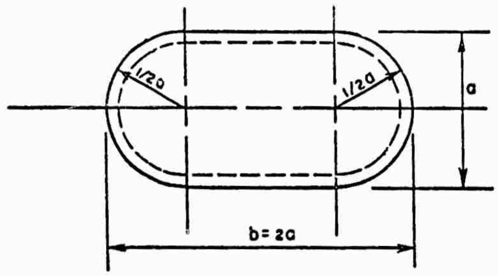
It should be recognized that one of the most critical factors affecting performance potential of large ocean-going GEM's is the operating height above the water. At this time, however, it is not known just what operating heights will be required for the ocean-going GEM because the performance of large GEM's operating at high speeds in various sea states is not yet well understood. It is axiomatic that the greatest economies can be obtained by the GEM when it operates very close to the surface. The lower the operating height the lower will be the power requirement and, consequently, the "hull" weight, initial costs, and operating costs. On the other hand, it is likely that very low operating heights, say the equivalent of 5 to 10 feet above still water for large platforms, would not be adequate to assure the practicability of high-speed operation in most situations. Operating height has been treated parametrically in the performance curves described in detail in Appendix D. Also, the influence of operating height on maximum speed and payload achievable in operational GEM's in the time frame of concern here is indicated in the discussions below.

Operating Height and Power Requirements

The GEM is supported on a cushion of air maintained between the base of the platform and the surface. Cushion pressure supporting the vehicle is achieved by high-volume fans and by a continuous annular jet at the periphery of the platform that entraps the air to maintain cushion pressure. The effectiveness of the annular jet in maintaining cushion pressure and minimizing power requirements depends critically on the height of the platform and on such factors as the arrangement of the jets and the jet discharge angle.

For practical designs, the cushion pressure achievable is a matter of only a fraction of a pound per square inch of base area and, as a consequence, the GEM requires a large base area to derive its lift. The larger the base area for a given cushion pressure and operating height, the greater is the lift augmentation and total lift derived. Thus, the efficiency of the GEM improves as the size of the platform is increased. In this study the performance curves have been projected on the basis of a lift-base area ratio of $L/S = 50$. (Lift equals total gross weight expressed in pounds, and base area is the total square feet of area enclosed by the annular jet.) An L/S ratio of 50 pounds per square foot appears to provide a realistic basis for examining the performance potential of the GEM.

From the standpoint of minimizing power requirements and minimizing efficiencies, the optimum plan form would be circular. This would not be well suited to operational employment of the platform. In this study the assumed plan form is as indicated in the sketch below where "a" equals the width or beam of the platform and "b" equals over-all length.



In Table II the plan form dimensions for displacements or gross weights up to 10,000 tons are shown, as are the horsepower requirements and the horsepower per ton of displacement. The greater efficiency of the larger GEM's is clearly indicated. The amount of power required decreases from 168 horsepower per ton at 500 tons displacement to about 70 horsepower per ton at 10,000 tons displacement. Note, however, that power requirements have been calculated for a 10-foot operating height and a cruise speed of 100 knots. At a 20-foot operating height and 100 knots, the power requirement of a 10,000-ton GEM would increase from 700,000 horsepower (shown in the table) to 1,000,000 horsepower (not shown in the table).

Detailed information on the basic assumptions and equations used in the calculations of performance potential described below is given in Appendix D.

Performance Potential

The maximum design payload that appears feasible for the GEM depends primarily on the size of GEM structure that can be built. Because of the space that would be available and the inherent flexibility available in location of engines and fans, it does not appear that power would be a primary constraint. As indicated earlier, a lack of basic information makes it difficult to estimate with a high degree of confidence the weight of structure required for seaworthy GEM's of large size. It is

Table II

PLAN FORM DIMENSIONS VERSUS TOTAL DISPLACEMENT
FOR GROUND EFFECT MACHINES

Δ (long tons)	L(10^6 lbs)	S($10^3 ft^2$)	a (ft)	b (ft)	HP[1] (000)	HP/ton Δ
500	1.12	22.4	78.2	156.4	80	168
1,000	2.24	44.8	112	224	140	134
2,000	4.48	89.6	158.5	317	210	106
3,000	6.72	134.4	194	388	280	94
4,000	8.96	179.2	224	448	350	87
5,000	11.2	224	251	502	410	82
6,000	13.44	268.8	274	548	470	78
7,000	15.68	313.6	296	592	525	75
8,000	17.92	358.4	317	634	580	73
9,000	20.16	403.2	336	672	640	71
10,000	22.4	448	354	708	700	69

1. The horsepower calculations assume a hovering height of 10 feet and a cruising velocity of 100 knots.

not known just what the structural limit on size might be and what effect sea conditions might have on the maximum feasible structural size. The data in Table II indicate that a 10,000-ton platform might well have a beam of over 350 feet and a length exceeding 700 feet. The construction of a platform of this size from lightweight materials capable of withstanding the structural stresses that will result in passing over an uneven surface may pose some difficult design problems. At this point in time there would appear to be a critical requirement for the construction of a fairly large (several hundred tons) experimental ocean-going GEM. Such a GEM would be a major requirement in a development program in providing basic data from which to extrapolate design criteria for large operational platforms.

In addition to the maximum size of structure, the type of power plant (lightweight gas turbine plant or lightweight nuclear plant) and the design operating height are critical factors to consider in assessing the payload potential of vehicles that are feasible within the time period of interest in this study.

In Fig. 19 are shown the maximum payload curves for speeds of 60 to 100 knots for both nuclear and non-nuclear powered GEM's. The top four curves show payload potential for GEM's with a total displacement or gross weight of 10,000 tons, if it is judged that structures of this size would be feasible to build. The lower set of curves shows payload potential assuming that the maximum-size GEM would be constrained to a total displacement of 3,000 tons. It is largely a matter of judgment but it has been concluded that the 3,000-ton GEM would probably be feasible, but that the feasibility of the 10,000-ton GEM would be questionable in the time period specified.

A 3,000-ton GEM would in itself be a large structure under the assumptions made in this study. It is seen below in the comparison of the size of a 3,000-ton and 10,000-ton GEM that the beam of a 3,000-ton GEM would be about 200 feet and the length almost 400 feet.

	3,000-Ton GEM	10,000-Ton GEM
Length	338	708
Beam	194	354

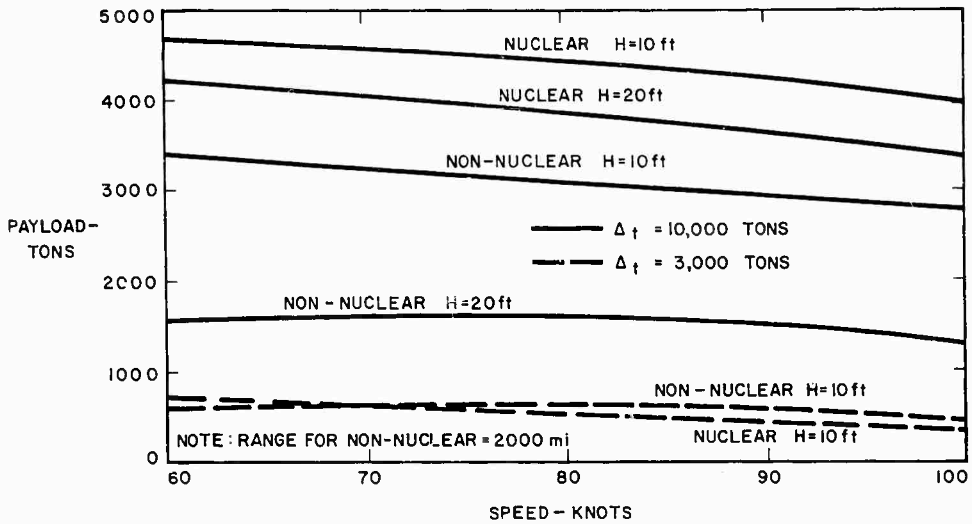


FIG. 19 GEMS
MAXIMUM PAYLOAD VERSUS SPEED
(Constrained by a Limit of 3,000 or 10,000 Tons
Total Displacement (Δ_t))

Seaworthy construction of vessels of this size using lightweight materials (aluminum) will demand extreme weight-consciousness if structural weight is not to exceed 25 percent of total displacement, as assumed as the basis for the projections of performance potential in this study.

In Fig. 19 it is seen that nuclear power is most advantageous for the larger payload platforms. At a 10-foot height the nuclear GEM offers payload potentials of from 4,000 to 4,500 tons, depending on speed. Both at a 10-foot and a 20-foot operating height, the payload potential of the nuclear GEM's exceeds that achievable by non-nuclear powered GEM's. The primary reason for this is the extremely high fuel requirement for the gas-turbine GEM's. At the smaller displacement (maximum of 3,000 tons) it is seen that payload potentials for the 20-foot operating height are practically zero. At the 10-foot operating height, payloads of 400 to 600 tons are feasible, depending on speed and type of power plant. The nuclear GEM's would, of course, have unlimited range. The design range of the non-nuclear GEM's shown in the chart is 2,000 miles. It is readily apparent that exploitation of the potential of the GEM is dependent on experimental and development work to demonstrate the design requirements and feasibility of very large GEM structures.

In the left-hand graph on Fig. 20 are shown basic relationships among speed, horsepower, payload, and total displacement (or gross weight) of GEM's at a 10-foot design operating height. A gas turbine power plant is assumed. It is seen that payload potentials of about 3,000 tons are feasible if a maximum size of 10,000 tons is granted. For smaller payloads, the size and horsepower required tend to increase only slightly with increases in design speed. At higher payloads, however, it is noted that total displacement and horsepower both increase sharply as the design speed of a GEM of a given payload is increased.

On the right-hand side of Fig. 20 is a comparable graph assuming an operating height of 20 feet. Note that there is a limit of about a 1,500-ton payload potential for a GEM of 10,000 tons total displacement. The increased horsepower required for GEM's of comparable payload at the higher operating height may also be seen. For example, at a 10-foot height a GEM with a 1,000-ton payload capacity and a speed of 100 knots would require about 400,000 horsepower (Fig. 20); at a 20-foot height the power requirement would increase to 900,000 horsepower. In both these charts, range is 2,000 miles. Comparable charts for other ranges are included in Appendix D.

The sensitivity of achievable payload potential to range requirements is illustrated in Fig. 21. This chart shows payload and other weight components as percentages of total displacement for GEM's powered

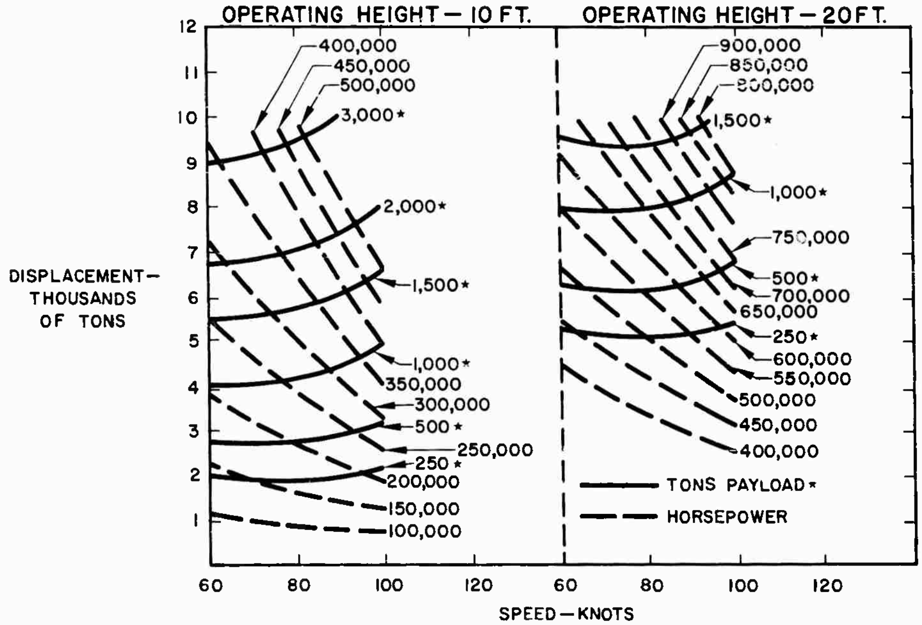


FIG. 20 GEMS
DISPLACEMENT AND PAYLOAD VERSUS SPEED AND
REQUIRED SHAFT HORSEPOWER, GAS TURBINE POWER PLANT
(Range, 2,000 Nautical Miles)

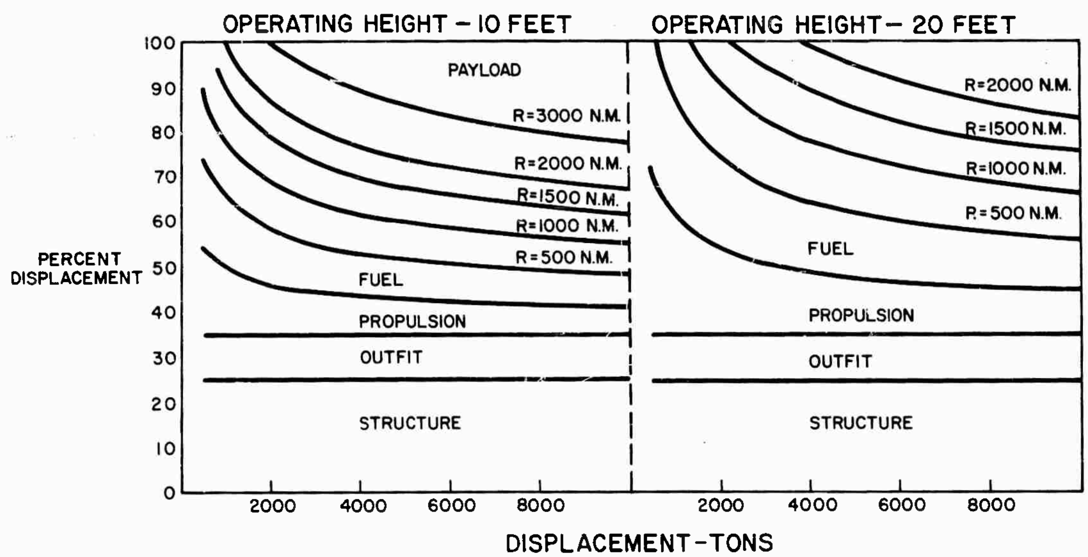


FIG. 21 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Speed, 80 Knots; Various Ranges)

by lightweight gas-turbine propulsion systems. The figure clearly shows the larger GEM's to be much more attractive, if such large structures are feasible. If the GEM is constrained to a 3,000-ton size, it is seen that payload potential is extremely limited unless design range is very short. The left-hand graph on this figure is for GEM's having a 10-foot operating height and a design speed of 80 knots. The influence of operating height may again be seen in comparing the above graph with the graph on the right-hand side, which assumes a 20-foot height and the same 80-knot speed. Range and payload potentials are very severely restricted by comparison with those feasible for GEM's with a 10-foot operating height.

In view of these limitations, the feasibility of nuclear power is considered, and Fig. 22 shows payload potential and other weight components for nuclear powered, 80-knot GEM's of various operating heights from 8 to 20 feet. For larger GEM's, say 5,000- to 10,000-ton total displacement, it is seen that payload potentials are very attractive. For smaller sizes, however, the high relative weight of nuclear shielding, again as for other concepts, makes nuclear propulsion less attractive. It is of significance to note also that the penalty associated with the higher operating heights is of much less significance for larger GEM's than for smaller GEM's.

Ocean-going GEM's appear to offer a significant potential as large platforms with speeds of up to 100 knots or so, particularly if nuclear powered. As indicated, however, there may be limits on the size of GEM's feasible in the projected time period, since specific structural requirements or design criteria have yet to be thoroughly investigated or understood. If large, high-speed platforms are in fact found to offer great potential for amphibious operations of the future, there is then an immediate requirement for intensified model and prototype development and testing to provide a basis for preliminary design of full-scale operational platforms.

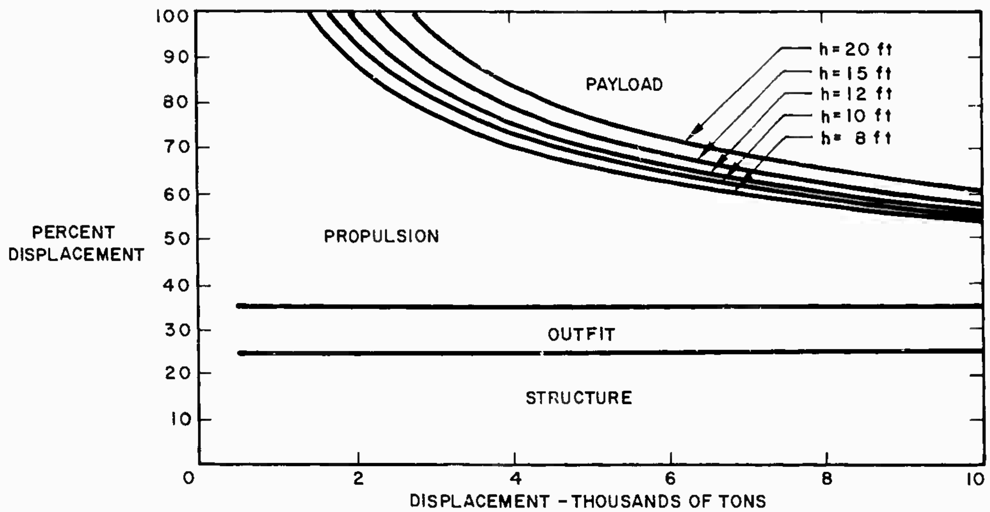


FIG. 22 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS
AS PERCENTAGE OF FULL LOAD DISPLACEMENT
NUCLEAR POWER PLANT
(Various Operating Heights: Speed, 80 Knots)

VII SUBMARINES

The feasibility of submarines ranging in payload potential from 1,000 to 40,000 tons and speeds of 20 to 40 knots or more has been investigated. The upper limit of the range of speeds considered in this study would afford some increase in speed over the speed of existing naval vessels; however, in contrast to all the other platform concepts investigated, the primary interest in the submarine relates not to high speed but to the capability for evasion or deception. At the same time it is to be noted that the submarine may be advantageous in providing modest increases in speeds without inordinate increases in power requirements. This is because the submarine hull characteristically has less wave resistance than a displacement hull of the same size.

High-performance submarines with payloads of less than 1,000 tons could also be employed in amphibious fleet operations, particularly when acting as convoy escorts. Submarines of this weight class, however, have not been investigated. There has been some discussion of the possibility of using semisubmersibles as a means of achieving more advantageous hull forms and reducing wave resistance. Possible arrangements of such hull forms are discussed and illustrated in the Technical Notes at the end of the text. The semisubmersible would not offer the full potential for reduction of wave resistance that the submarine forms would, since it would be sensitive to surface motions and storm conditions. Also, it would require submarine-type construction (water-tight, limited-pressure hull design), which leads to higher costs than those associated with surface vessels, but would not provide the advantages of true submarines. Operation in harbor areas could be as restricted, or more restricted, for semisubmersibles than for true submarines, depending on the performance and controls designed into the semisubmersible. In view of these considerations, it was concluded that the potential advantages of semisubmersible hull forms did not warrant further investigation, and that attention should be focused on true submarines.

In this investigation of submarines it has been assumed that nuclear propulsion would be of primary interest. Two types of reactor systems were investigated: the present-day pressurized water reactor system and an advanced reactor system requiring 50 percent less weight and space than present systems. A gas-cooled reactor offering such savings in weight and space would appear to be feasible in the time period of concern here; such systems would require less development than the much lighter-weight nuclear plants that it was concluded could be developed for propulsion of hydrofoils, planing hulls, or GEM's. Power plant weight is not as critical for submarines of limited operating depth as for other platform

concepts, since propulsion plant weight is not a major percentage of the total weight of the size of submarines under consideration.

Hull Forms and Power Requirements

A ship running under water with sufficient submergence encounters no wave-making resistance. Submerging a vessel, however, increases its wetted surface and, at low speeds, the resultant increase in frictional resistance more than offsets the absence of wave resistance. At high speeds, the submarine does in fact offer decided advantages over a displacement hull of comparable size in terms of less total resistance. On the debit side, it should be noted that thc submarine hull must resist water pressure and, because of structural considerations, it will accordingly have a greater displacement than a surface vessel with the same payload.

The submarine hull having a body-of-revolution shape or circular cross section has the minimum resistance. At large payloads and displacements, the large diameter of these hull forms will create undue difficulties in loading, unloading, and drydocking. Eighty feet has been assumed to be the maximum practical diameter. An alternative to this low-drag form is a rectangular cross section with which volume is obtained by maintaining a given maximum allowable draft and increasing beam and length. Thirty-six feet was taken as the maximum draft. Structures are a more difficult design problem in the rectangular cross section and, for very large submarines, this hull form results in excessive dimensions and power requirements.

Submarine design and operational problems are considerably affected by the nature and the density of cargo or weapons systems to be carried. The lower the cargo density, the larger the submarine must be. Performance curves in this study have been presented on the basis of payload weight potential; in practice, volumetric limitations for certain types of cargoes would not permit realization or utilization of full weight capacity. The submarine configured as a dry-cargo transport, particularly for larger equipments such as vehicles and unit loads of supplies, presents an extremely difficult problem in hull arrangement and structural design. In addition to large, clear deck areas, large openings must be provided in the submarine's pressure hull to provide cargo access, thus creating a difficult engineering problem. Difficulties in the design of dry-cargo submarines would relate not to major technological limitations but to such factors as design complexity, cost, and inefficient utilization of space. The payload potential of a dry-cargo submarine would be only about 50 percent of the payload potential of a tanker submarine of the samc

displacement. This is because of the increased structural complexity and the increased space required for variable ballast tanks in non-tanker configurations.

In this study, submarines of both circular and rectangular cross section have been considered, and data have been developed on both dry-cargo and liquid-cargo, or tanker, configurations. The submarines under consideration here are intended to operate only at depths up to 400 feet (well below the critical depth for elimination of wave-making resistance). However, the pressure hull realistically has been designed (weight allowances have been made) for somewhat greater depths (test depth of 1,000 feet) as a safety factor to permit recovery from momentary loss of control.

Horsepower requirements in this study were derived as a function of displacement and speed, as follows:

$$SHP = KD^{2/3}V^{3}$$

where

SHP = shaft horsepower

D = submerged displacement (1.1 times surface displacement)

V = maximum submerged speed

K = constant, depending on proportions and appendages of the submarine hull.

A value of K = .0032 was adopted for submarines with rectangular cross sections and a value of K = .0022 for circular cross-sectional submarines. Further discussion of the basis for the projections of power requirements and weight breakdowns for submarines is given in Appendix E.

Performance Potential

In examining the speed and payload potential of the submarine, it is found that the submarine has significantly lower power requirements relative to the surface displacement hull for a platform of given payload and speed. For example, the power of a 45-knot, rectangular cross section submarine with a payload potential of 5,000 tons (dry cargo) would be 300,000 horsepower. Its total surface displacement would be about 31,000 tons. A 45-knot nuclear propelled, steel hull displacement vessel of the same payload potential would require well over 400,000 horsepower. Total displacement would be on the order of 15,000 tons. For a mild steel hull displacement vessel propelled by a geared turbine power plant of the

same speed and payload, the power required would be about 600,000 horsepower and total displacement would be about 27,000 tons. Range would be limited to 2,000 miles. (See Figs. 4 and 8 for displacement hull characteristics and Fig. 25 for the submarine characteristics.) The comparative power advantage of the submarine would be even greater in a tanker configuration. A rectangular cross section tanker with a 5,000-ton payload and 45-knot speed would require only a little over 200,000 horsepower and would have a total displacement of only 18,000 tons (see Fig. 24)

At up to 45 or 50 knots, horsepower requirements would not appear to limit the maximum payload achievable in a submarine. No definite physical constraint or limit on the maximum payload potential has been established. One of the characteristics of the submarine is its potential for extremely high payloads, particularly for liquid-cargo transport where problems of hull configuration, layout, and design complexity would be minimized.

In Fig. 23 is shown the payload potential achievable in a dry-cargo submarine, assuming size is constrained to a maximum surface displacement of 25,000 tons, 50,000 tons, or 75,000 tons. It is assumed that the submarine would have an advanced reactor system and a hull with a rectangular cross section. Draft would be limited to 36 feet for harbor entry. A 75,000-ton submarine, such as is shown in the figure, would have an over-all length of 750 feet and a beam of 120 feet (see Appendix E). A submarine of this size is considered feasible within the near-term state-of-the-art. If a more conservative view is taken, limiting the maximum size to, say, 50,000 tons (a length of 625 feet and beam of 95 feet), payload potentials of 8,000 tons at 45 knots or payloads up to 18,000 tons at 20 knots would be achievable. With the same constraints on maximum over-all size or displacement, somewhat higher payload potentials could be achieved in submarines having circular cross sections and tanker configurations. However, it is considered that primary interest here is in submarines of rectangular cross section because this hull form will minimize draft requirements (important because of the relatively shallow waters in port areas and possible requirements to operate close inshore) and is better suited to internal layout for dry-cargo use.

In Fig. 24 are shown basic relationships among speed, horsepower, payload, and total surface displacement for submarines having rectangular cross sections, powered by advanced reactor systems, and configured as liquid-cargo carriers, or tankers. The curves have not been extrapolated beyond 45 to 50 knots because it is judged that this is probably the practical limit on speed. This judgment has been made from the standpoint of practical design for good control and maneuverability, particularly for larger submarines. Experimental work may prove higher speeds to be feasible.

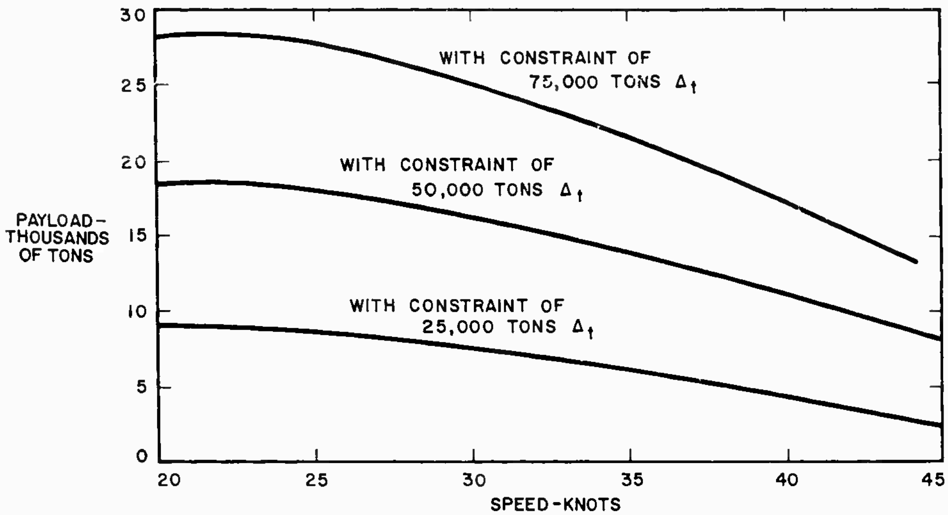


FIG. 23 SUBMARINES, MAXIMUM PAYLOAD VERSUS SPEED, RECTANGULAR CROSS SECTION, DRY CARGO, ADVANCED REACTOR
(Constrained by a Limit of 25,000, 50,000, or 75,000 Tons Total Displacement (Δ_t))

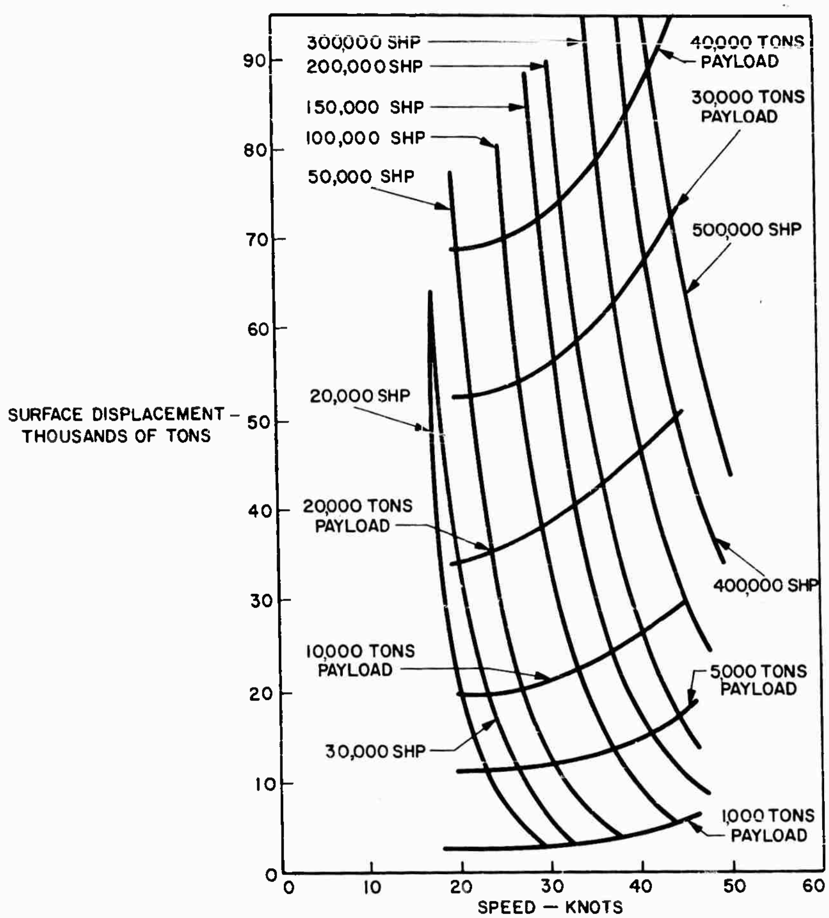


FIG. 24 SUBMARINES, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED HORSEPOWER, RECTANGULAR CROSS SECTION TANKER, ADVANCED REACTOR

Figure 25 shows data on dry-cargo submarine configurations comparable to those shown in Fig. 24 for tanker configurations. In both cases rectangular cross sections and advanced nuclear reactors are assumed. The loss in efficiency of space utilization with the dry-cargo configuration may be seen in comparing the two charts. For example, a 10,000-ton payload tanker with a 45-knot speed capability would require a 30,000-ton total surface displacement and 300,000 horsepower. A 10,000-ton payload dry-cargo submarine of the same speed would require a total surface displacement of 57,000 tons and about 450,000 horsepower. Comparable charts for submarines with circular cross-sectional hulls and powered by pressurized nuclear reactors (present types) are given in Appendix E.

Figures 26 and 27 are comparable charts indicating the payload potential and major weight breakdown as percentages of surface displacement for tanker submarines (Fig. 26) and for dry-cargo submarines (Fig. 27). Again, these charts assume rectangular cross sections and advanced reactors. Displacements up to 90,000 tons are given, along with speeds of 20 to 40 knots. These charts indicate clearly that propulsion system weight, which increases somewhat with higher design speeds, is not as critical in submarine design as in other platform concepts.

All of the potential performance charts described here have assumed that advanced, gas-cooled nuclear reactors (one-half of the weight of present pressurized water reactors) would be employed. Such reactors are not now available, but it has been judged that they could be developed and made available within the time period of interest in this study. In point of fact, use of existing pressurized water reactors would not sacrifice greatly the payload potentials achievable in future submarine designs, since it is feasible to build submarines of very large total displacement. However, use of larger hulls would greatly increase costs and, as will be seen, submarine construction costs are high relative to those for alternative platform concepts.

It is of some significance to note here that submarine technology is highly developed. While submarines of the size and speed projected as being feasible are not now in being, it is considered that the major effort required to achieve the potentialities described would be in engineering and design work rather than in more basic research, development, or experimental programs. Existing submarine technology would permit immediate construction of larger, faster submarines, but construction of large GEM's, on the other hand, would be highly dependent on intensive developmental work and construction of comparatively large experimental GEM's to advance GEM technology and to establish basic design criteria.

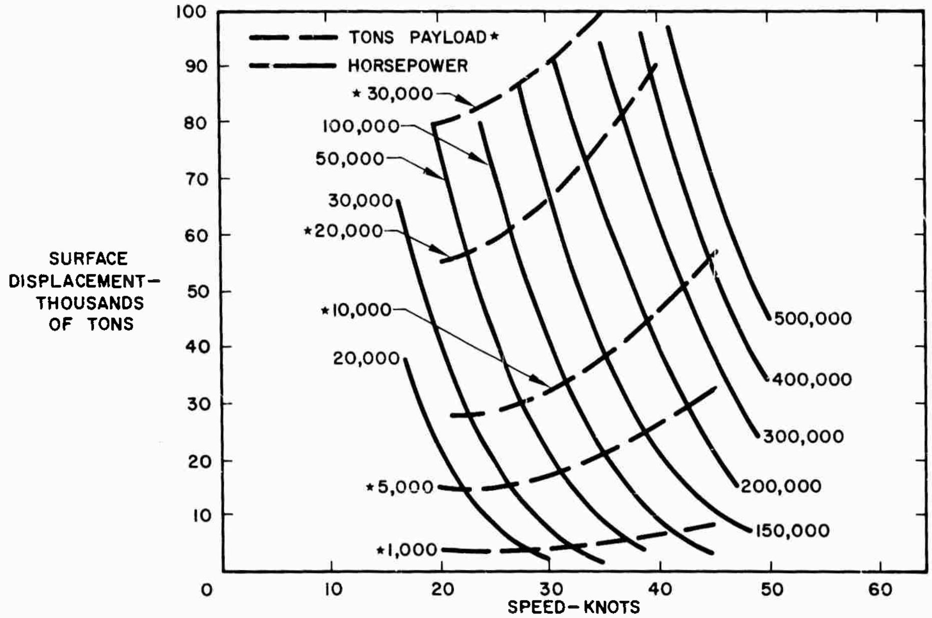


FIG. 25 SUBMARINES, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED HORSEPOWER, RECTANGULAR CROSS SECTION DRY CARGO, ADVANCED REACTOR

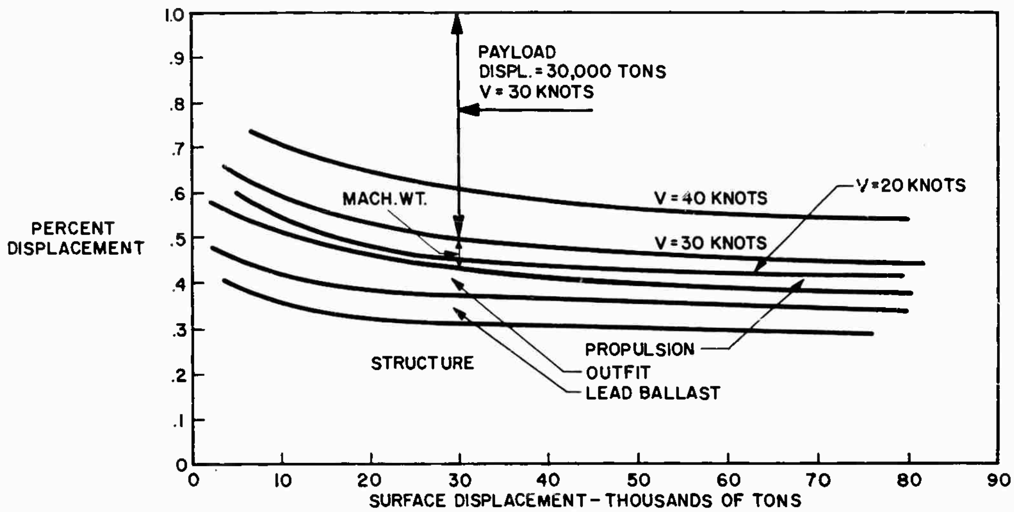


FIG. 26 SUBMARINES, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF TOTAL DISPLACEMENT VERSUS DISPLACEMENT RECTANGULAR CROSS SECTION, TANKER, ADVANCED REACTOR

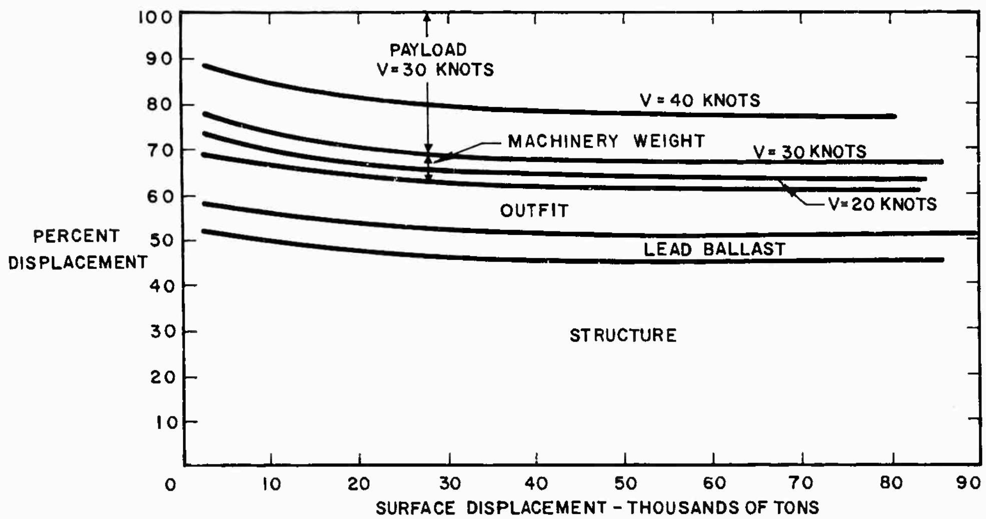


FIG. 27 SUBMARINES, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF TOTAL DISPLACEMENT VERSUS DISPLACEMENT RECTANGULAR CROSS SECTION, DRY CARGO, ADVANCED REACTOR

VIII COMPARISON OF ALTERNATIVE PLATFORM CONCEPTS

It was indicated in Section I that the basic assessment of the comparative suitability of alternative platform concepts as major fleet units for the amphibious task force of the future could not be made in this study but would, of necessity, have to be accomplished within the framework of the larger ONR study of weapons systems for future amphibious operations. Nevertheless, it is of interest here to compare the relative feasibility and probable construction costs of alternative platform concepts and to consider the relative sensitivity of the various concepts to adverse sea conditions.

Comparative Power Requirements and Probable Construction Costs

A series of charts has been developed to show, for specified payload equivalents, the speed potential of each alternative platform concept. Each specified payload equivalent is presented on a single chart, on which two graphs are shown side by side. The first graph shows total horsepower as a function of speed, plotted for each platform alternative feasible at the specified payload. The second graph shows capital costs as a function of speed, plotted for each alternative. The charts can be studied to ascertain the platform concept that (1) would minimize power requirements for a given speed and payload equivalent (thus, comparative power requirements are used as one relative measure of merit) and (2) would minimize initial cost for a platform of given speed and payload potential. Moreover, these charts may be studied to ascertain the speed range to which each alternative concept is best suited.

The horsepower requirements have been derived from the potential performance and power curves described in previous sections. Comprehensive sets of performance curves for each alternative platform and propulsion combination are presented in the appendixes. It will be noted that range is not constant among the alternatives plotted on a single chart. Nuclear-propelled platforms would have essentially unlimited range. The range in all other cases is 2,000 miles, unless a shorter range is specified on the note at the bottom of the chart. Shorter ranges occur where the specified payload is not achievable at the basic 2,000-mile range.

The estimates of capital costs or initial construction costs have been calculated on the basis of hull size, hull material, installed horsepower, and type of propulsion. As presented in Table III, the cost equations have been derived by taking into account characteristic

Table III

EQUATIONS FOR ESTIMATING COMPARATIVE CAPITAL COSTS OF ALTERNATIVE PLATFORM CONCEPTS

Concept	Equation for Cost in Dollars
Displacement Hulls	
1. Non-nuc, steam turbine, steel	$2{,}500\ (\triangle) + 681(SHP)^{0.865}$
2. Nuclear, steam turbine, steel	$2{,}500\ (\triangle) + 681(SHP)^{0.865} + 85(SHP)$
3. Non-nuc, gas turbine, aluminum	$2{,}700\ (\triangle) + 65(SHP)$
4. Nuclear, gas turbine, aluminum	$2{,}700\ (\triangle) + 150(SHP)$
Planing Hulls	
1. Non-nuc, gas turbine, aluminum	$12{,}000\ (\triangle) - 1.50\ (\triangle)^2 + 65\ (SHP)$
2. Nuclear, gas turbine, aluminum	$12{,}000\ (\triangle) - 1.50\ (\triangle)^2 + 150(SHP)$
Hydrofoils	
1. Non-nuc, gas turbine, aluminum	$29{,}160\ (\triangle) - 4.30\ (\triangle)^2 + 75\ (SHP)$
2. Nuclear, gas turbine, aluminum	$29{,}160\ (\triangle) - 4.30\ (\triangle)^2 + 160(SHP)$
GEM's	
1. Non-nuc, gas turbine, aluminum	$17{,}000\ (\triangle) - 0.85\ (\triangle)^2 + 65\ (SHP)$
2. Nuclear, gas turbine, aluminum	$17{,}000\ (\triangle) - 0.85\ (\triangle)^2 + 150(SHP)$
Subs (Rectangular and Circular)	
1. Tanker, pressurized water reactor	$4{,}200\ (\triangle) + 700(SHP)$
2. Tanker, advanced gas-cooled reactor	$4{,}200\ (\triangle) + 200(SHP)$
3. Dry cargo, advanced gas-cooled reactor	$6{,}500\ (\triangle) + 200(SHP)$

Note: $(\triangle)$ = full-load displacement.
SHP = shaft horsepower.

Source: M. Rosenblatt & Son, Inc., Naval Architects and Marine Engineers.

differences in costs of hull materials (steel and aluminum), differences in costs of fabrication techniques required, and differences in probable costs of steam turbine, gas turbine, and nuclear propulsion systems. It should be recognized that these cost equations provide only preliminary estimates of the comparative capital costs of alternative platforms.

The cost estimates could not be based on detailed analysis of projected bills of materials, specific fabrication jobs, and assessment of labor requirements, since such estimates would, of course, be dependent on the availability of at least preliminary design layouts and configuration studies. Nevertheless, the cost equations are considered valid as a means of developing an appreciation of the probable differences in capital costs of basically different platform concepts. No attempt has been made in this study to assess the probable relative costs of developmental work, including model and prototype design and testing, that would be required to achieve the projected design potentials of each alternative. Moreover, no estimate has been made of operating costs. The different platform concepts would have significantly different operating costs, even platforms of comparable speeds.

The comparative charts showing horsepower and probable capital costs of alternative platforms as functions of design speed are presented as Figs. 28-30. Figure 28 represents a 250-ton payload equivalent, Fig. 29 a 2,000-ton payload equivalent, and Fig. 30 a 5,000-ton payload equivalent. The charts are illustrative of the types of comparisons that can be made using the characteristics and performance data developed in this study. The significance of the comparisons of the three specified payload equivalents (Figs. 28-30) can be appreciated only through careful study of the charts; however, a number of general observations of some significance are in order here.

In Fig. 28 it is noted that all of the platform concepts (except the submarine) appear.[1] This figure is for a payload potential of only 250 tons, which is probably of marginal interest in this study. The basic reason for its inclusion here is that certain of the concepts are limited in potential to such small payload equivalents. It will be noted, for example, that neither the planing hull nor the hydrofoil appears on Fig. 29, which shows the horsepower and cost characteristics of platforms having a 2,000-ton payload equivalent.

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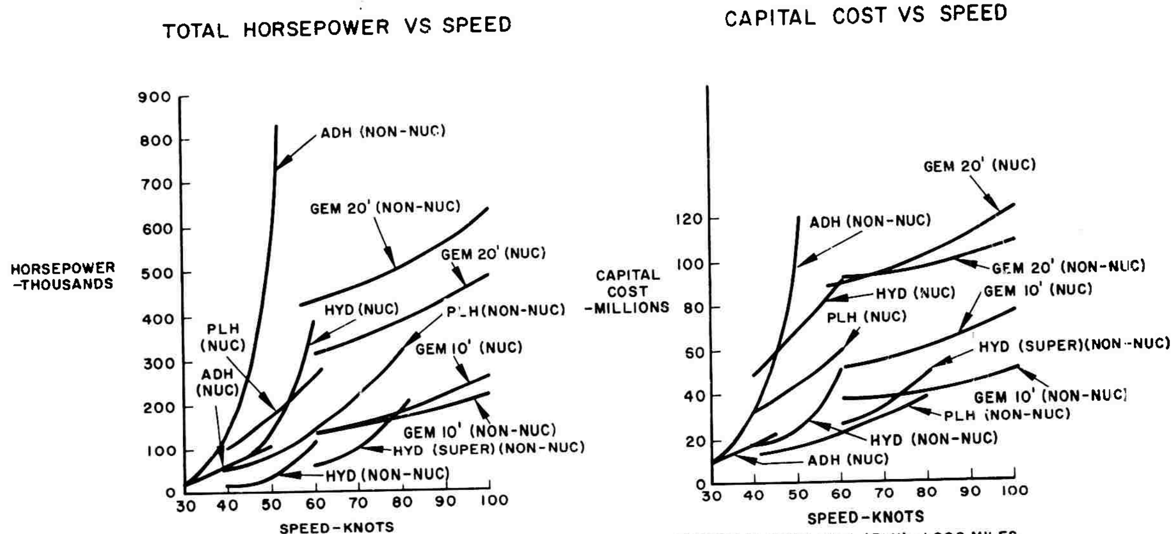
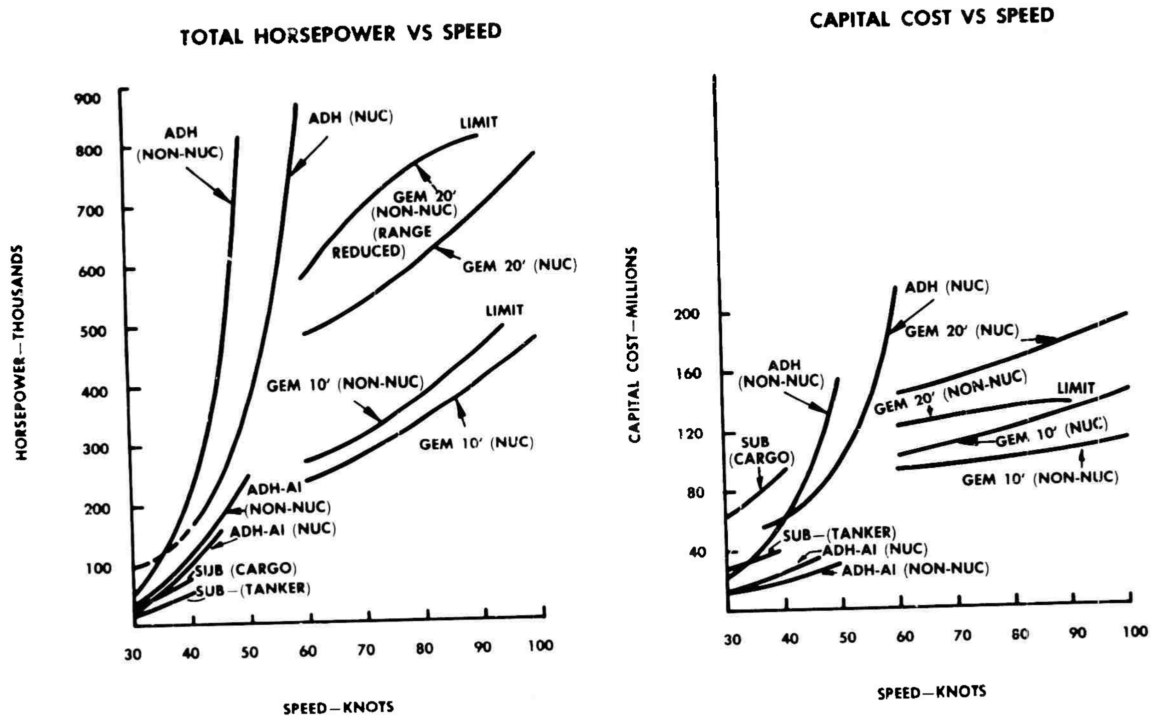
1. No performance data were developed for submarines with payload potentials under 1,000 tons.



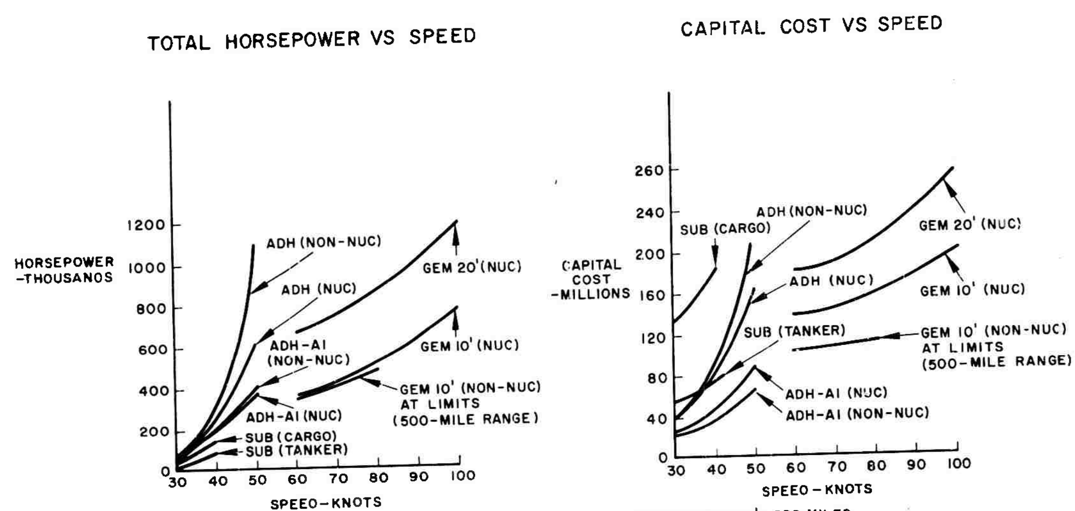
NOTE: RANGE OF NON-NUCLEAR PLATFORM IS 2,000 MILES EXCEPT: PLANING HULL (PLH)-1,000 MILES HYDROFOIL, SUBCAVITATING (HYD)-1,000 MILES, HYDROFOIL, SUPERCAVITATING (HYD SUPER)-500 MILES

FIG. 28 HORSEPOWER AND COST VERSUS SPEED OF ALTERNATIVE PLATFORM CONCEPTS
250-TON OR DE PAYLOAD EQUIVALENT
(Range not Constant)



Note: Range of non-nuclear platforms is 2,000 miles except:
GEM 20' - 1,500 miles.

FIG. 29 HORSEPOWER AND COST VERSUS SPEED OF ALTERNATIVE PLATFORM CONCEPTS
2,000-TON OR LST AND CG PAYLOAD EQUIVALENT
(Range not Constant)



NOTE: RANGE OF NON-NUCLEAR PLATFORM IS 2,000 MILES EXCEPT: GEM 10' - 500 MILES

FIG. 30 HORSEPOWER AND COST VERSUS SPEED OF ALTERNATIVE PLATFORM CONCEPTS 5,000-TON OR AKA AND AF PAYLOAD EQUIVALENT (Range not Constant)

For the 250-ton payload equivalent (Fig. 28) it may be noted that, from the standpoint of minimum power requirements, the non-nuclear hydrofoils (subcavitating and supercavitating) look most attractive up to a speed of 80 knots. Cost-wise, the hydrofoils are somewhat more expensive than the planing hull. The planing hull, as has been noted, would be restricted to favorable sea conditions. Above 80 knots the non-nuclear GEM having a 10-foot operating height looks most attractive, considering both power requirements and costs.

In Fig. 29 are the comparisons for the 2,000-ton payload equivalent. Submarines are included in these comparisons. It is found that, from a power standpoint, they appear quite favorable up to speeds of about 40-45 knots, as already noted. The costs of submarines are seen to be higher than the costs of both nuclear and non-nuclear powered advanced displacement vessels with aluminum hulls. The cost of the dry-cargo submarine is seen to be about twice that of the tanker submarine because it is a much larger vessel and has a much higher horsepower requirement. The aluminum hull displacement vessels look extremely favorable in their speed regime as compared with steel hull advanced displacement types in terms of both power required and costs. Above 50 to 60 knots, the GEM looks most favorable; the GEM with the 10-foot operating height is significantly less costly than the GEM with the 20-foot height, as would be expected.

Figure 30 shows the comparative power and cost of alternative platform concepts with a 5,000-ton payload equivalent. The comparisons are similar to those observed for platforms with a 2,000-ton payload. From a cost standpoint, advanced displacement hulls of aluminum construction look very favorable up to 50-60 knots. The non-nuclear aluminum displacement hull is only slightly less costly than the nuclear powered displacement hull of aluminum construction. Nuclear propulsion would, of course, afford unlimited range.

The figure indicates clearly the advantage of nuclear propulsion for the larger GEM. The non-nuclear GEM with a 10-foot height has an extremely limited range potential, and its design speed is limited to 80 knots. At higher speeds a 5,000-ton payload equivalent cannot be obtained in the non-nuclear GEM without further degradation of range. The non-nuclear GEM with a 20-foot operating height simply is not feasible for payloads of 5,000 tons, as indicated by its omission from the chart. Again, this figure illustrates the very high comparative costs of the dry-cargo submarines.

Performance under Adverse Sea Conditions

One of the basic objectives of the investigation of new platform concepts for amphibious operations is the attainment of increased speed. A prime consideration in this area is the ability of each vehicle to maintain its design speed as sea conditions deteriorate. Because ocean-going platforms are under discussion, it becomes important to determine the performance characteristics of the various alternatives in open-sea operations.

In general, seakeeping problems arise from the turbulence encountered on the interface between the water and the air and, for the surface vessel, from inadequate stability over expected operating conditions. Submersible craft attempt to maintain speed by operating below this turbulence, while displacement hull technology is directed toward better ways to operate in this zone. For the hydrofoil and the GEM the problem is lessened by raising all or most of the hull above the turbulence. In all these concepts, as speed increases the difficulties mount. Slamming into a wave crest at high speeds requires a rugged structure for GEM's, hydrofoils, displacement hulls, and planing hulls. The tendency of displacement hulls to "work" in a seaway results in a considerable loss of speed due to increased resistance and through power reductions to reduce ship motion.

There is very little quantitative information available on speed degradation in adverse seas, even for existing types of vessels.[1] The great preponderance of estimates of sea state results from visual observations, and these are subject to some question.[2] Furthermore, the willingness of ships' captains to operate at high speeds varies with the individual skipper. Thus, the discussions below depend on relative considerations rather than on precise data as to human or structural limitations.

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1. The David Taylor Model Basin (DTMB) is undertaking a four-year program designed to produce definitive, quantitative information on displacement hull speed degradation.
2. In a recent example, the crew of a test vehicle reported encountering waves 3 to 4 feet high in the test area. Coast Guard observations, using instruments, showed that the actual wave heights were between 1.5 and 2 feet.

Displacement Hulls

Displacement hulls operate in the turbulent surface of the sea. Hence, they are subject to the lateral motions of roll, yaw, and sway and the longitudinal motions of surge, heaving, and pitching. Of these six, the most serious are heaving and pitching, or the vertical translational and rotational motions. These result in large vertical motions which cause shipping of water, slamming, and propeller racing and loss of efficiency. Furthermore, as other motions of the ship are controlled (through roll stabilizers, for example) heaving and pitching become more noticeable.

Present displacement hulls (destroyer size and up) experience some speed loss in a state 4 sea (see Technical Notes at end of text for description of sea states), serious speed degradation in an advanced state 5 sea, and a mandatory power reduction in a state 6 sea. Present displacement hulls have the capability to ride out all but the most extreme sea conditions.

The head sea is the most difficult for displacement hull operations, primarily because the heaving and pitching motions are most severe in this environment. Displacement hulls experience a 15-25% speed loss when heading directly into a state 5 sea with no reduction in power. A power reduction soon becomes mandatory, however, or serious damage can result, even to very large ships. For example, the heavy cruiser USS Pittsburgh (17,200 tons full-load displacement) lost 100 feet of her bow during a typhoon in the western Pacific on June 5, 1945. Generally speaking, ships' captains avoid a well-developed head sea if at all possible.

A bow sea is preferable to a head sea, and ships will often "tack" into a sea, taking it on alternate bows in order to make a good line of advance. Motions, and therefore speed degradation, are somewhat reduced, although a state 6 bow sea can present serious problems to a vessel.

A beam sea magnifies the rolling motion. As such, it is not desirable to take an advanced sea on the beam. On December 18, 1944, three U.S. destroyers were lost in a typhoon through capsizing. Most seamen have a fear of getting "in the trough" of a storm sea because of the danger of capsizing. Further, a long slim hull that operates well in a head sea is generally a very poor performer in a beam sea due to her long profile. Displacement hulls will generally avoid a beam sea of state 5 and above, and should not operate in a state 6 or above beam sea.

Characteristically, displacement hulls operate most effectively in quartering and following seas. In fact, it is not uncommon for a following sea to produce a better speed of advance than that indicated on

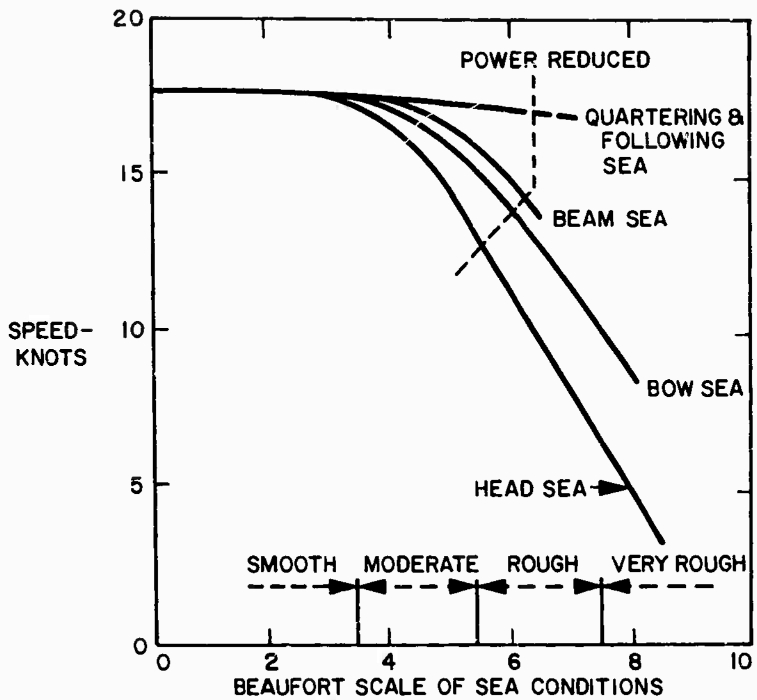
the engine room telegraph. A following sea results in some yaw or drift around the course established (sometimes 30 to 40 degrees) and also causes propeller racing. In severe cases, a giant following sea will cause a vessel to "fall off" into the trough or will raise the screw completely out of the water. Generally speaking, however, displacement hulls perform well in quartering and following seas.

The performance of displacement hulls in various seas is shown graphically in Fig. 31. This figure shows comparative speed reductions required by a Victory-class cargo ship as a function of sea state, assuming a quartering sea, beam sea, bow sea, and head sea. Other displacement vessels would show similar characteristics.

It is apparent from the study team's investigations that advances in displacement hull technology may significantly improve performance potentials in adverse sea conditions. Specifically, there are two areas of research that appear promising.

Since heaving and pitching are the most serious problems encountered by displacement hulls, work is under way to eliminate or reduce these motions. Quite by accident, DTMB discovered a promising avenue for research. A destroyer was fitted with new sonar gear, which necessitated the installation of a large sonar dome in the bow of the vessel. Sea trials showed the ship's motions to be materially reduced and the seaworthiness improved. Investigation showed that the increased seaworthiness was obtained by greatly decreasing the relative motion of the bow. DTMB indicates that some improvement can therefore be anticipated in existing displacement hull vessels.

Tests conducted in the tank at DTMB indicate that as the frequency of wave encounter is increased the ship's motions are reduced. In effect, the vessel does not follow the contours of the sea but rides on the higher parts of successive waves. This phenomenon is known as supercritical operation. For most displacement hulls and most sea conditions, supercritical operations occur somewhere between 30 and 40 knots. Tests indicate that a vessel in this condition will ride through a state 5 sea as though "she were on a railroad track." If the test program continues to be promising, high-speed displacement hulls such as described in this study should be able to take advantage of this condition and achieve good speed performance under rough sea conditions.



Source: Lewis, E.V., "Log Analyses," Experimental Towing Tank, Stevens Institute of Technology, Hoboken, N.J., 1958.

FIG. 31 TREND OF SPEED WITH SEA CONDITIONS FOR VICTORY CARGO SHIPS

Planing Hulls

The planing hull operates on the surface of the sea. Its salient characteristics are a flat bottom, comparatively shallow draft, and a reasonable payload and speed potential. Its most serious and persistent defect involves the design itself, since heavy slamming is encountered when operating in any kind of a seaway.

The head sea is the most serious condition for planing hulls, since slamming is harmful and sometimes dangerous to both men and machinery. High performance of large planing hulls would be out of the question in a state 3 sea or above. When "flying" off a wave crest, the hull presents a large sail area into the wind and, in some instances, there may be danger of capsizing.

Planing hulls tend to "work" in a beam sea, although they possess a great deal of lateral stability. The flat hull slides off waves, and it is difficult to maintain a steady course. There is also some danger of abruptly heading up into the sea, particularly at high speeds. A following sea also presents problems for a planing hull because of its inability to maintain a set course under these conditions. A sea from the quarter or astern will produce a considerable yaw, or "surfboarding" effect, and broaching may be a common occurrence with this type of hull.

Of the various concepts studied, the planing hull characteristically exhibits the poorest seakeeping qualities. It should be stated that this assessment of planing hulls as ocean-going platforms is based entirely on extrapolation of the performance characteristics of smaller planing hull configurations. It can be said with some confidence, however, that the seaworthiness of this hull form is not good. There is some possibility that experimental work on model or prototype ocean-going planing hulls could improve the present outlook.

GEM's

The ground effect machine offers the greatest potential speed increase over existing naval vessels of any of the concepts examined, being capable of 100 knots or more. GEM technology is still in an early stage, and only relatively small vehicles are presently under active development. The GEM operates free of, but very close to, the turbulent surface. Seaworthiness depends, therefore, on either (1) the ability of the GEM to keep high enough to clear the surface, possibly requiring it to "follow the sea," or (2) its ability to contact the surface only intermittently and to be able to withstand severe impacts at these times without damage to the vehicle or disruption of control.

Present GEM's, both here and abroad, are designed to operate in a state 3 sea and to survive in a state 6 sea. The cushion is reported to be able to flatten the crests of waves, while the vehicle has a limited ability to "knife through" some of the higher waves encountered. As noted, there are severe increases in power requirements as operating height is increased. Mr. Weiland of Douglas Aircraft has suggested that an efficient GEM could have 2 feet of clearance for each 100 feet of length. This height could likely be doubled, but only with considerable loss in efficiency. If these figures are correct, then a 10,000-ton ocean-going GEM, as described earlier, would have an effective operating height of 15 feet. For a 3,000-ton GEM, the comparable operating height would be less than 10 feet.

The performance of the GEM in open-sea conditions is not well understood, but some observations can be made. A head sea would be the poorest condition for the GEM. This would result from a tendency to contour the waves. When wave length is $1\frac{1}{4}$ to 2 times the length of the vehicle, there would probably be a marked tendency to "dive into" wave crests. In operating over a confused sea, the GEM is likely to encounter directional forces that would be difficult to counteract. In the displacement condition, a GEM would be a close cousin of the planing hull. Due to her speed when flying, the GEM would be susceptible to severe slamming when operating into a head sea unless it had very high operating height--above that suggested as being feasible for "efficient" GEM's.

The motion of the GEM would be a good deal less severe in a bow sea than in a head sea. The GEM should not have unusual difficulties running in a beam sea. The wind may well be a real problem, however, due to the large sail area of the GEM. Navigation and maintaining course could be difficult, and wind-blown spray might be another problem. There might also be some tendency to fall off and slide down the sides of the waves. It is likely that there would be some surfboarding in the GEM, with the usual danger of broaching. It is quite likely that the GEM would have an excellent speed of advance in these conditions, however, much as an aircraft benefits from a tail wind.

In brief, the seaworthiness of the GEM will depend in large measure on its ability to keep clear of the surface of the water. This means a large vehicle. As discussed elsewhere, large GEM size will depend on the satisfactory solution of structural and propulsive problems. The seaworthiness of GEM's may improve as more efficient or economic methods of maintaining operating heights are devised. The articulated curtain or "skirt" is now being investigated. Skirts of up to 15 feet are now considered feasible, but performance characteristics over rough seas are not known. Skirts present an added maintenance burden and could constitute

a heavy drag. Efforts are also under way to increase vehicle clearance by use of new techniques for improving efficiency in use of available power, such as in recirculation of cushion air.

Hydrofoils

The hydrofoil has the potential of being the most seaworthy of any of the concepts except submarines. Similar to the GEM, the hydrofoil depends for its seaworthiness on maintaining clearance between the surface and the bottom of the main part of the hull. As a displacement hull, the hydrofoil would be able to survive in advanced sea states because of the stability derived from the extended foils.

The struts of the hydrofoil may be theoretically extended to any desired length to improve seaworthiness, but stability and structures must be considered in practical vessels. The H.S. Denison was designed for sea state 3, although Grumman has run her in an advanced sea state 4 at 55 knots. It would appear that hydrofoils of 500 tons or larger will be able to operate in a state 6 sea with no difficulty. Of course, once the limiting foil clearance is reached, speed is very rapidly lost. It also appears that the hydrofoil will be capable of slicing through the tops of waves to some limited extent.

Head and bow seas are the area of greatest difficulty for hydrofoil craft. There is some tendency to contour, although this is reduced as speed is increased. The hydrofoil must avoid "diving into" the crests of succeeding waves. A surface-piercing foil system is able to resist this tendency, since lift is automatically augmented as the craft sinks in the water. With a submerged foil system, however, there is no built-in or natural compensating device, and there is a real danger of "diving" if sensor systems fail. Some sort of reliable sensing system is mandatory.

A beam sea is generally no problem in hydrofoils, but the situation changes as relative foil length is increased. As the length increases, the foilborne center of gravity rises, and the craft is less able to withstand lateral forces. There is also some danger of "ventilating" the foils. This occurs when one or more of the foils are raised out of the water by sea motions and results in immediate loss of lift. As will any ocean-going vehicle, hydrofoils will probably avoid a large beam sea if at all possible.

In tests to date, the hydrofoil has had little difficulty in running before the sea. There is some danger of ventilation, as discussed above, particularly with a submerged foil system. As far as is known, these craft are quite stable in a following sea, with little tendency to fall off or broach.

Submarines

It is not necessary to provide a detailed discussion of the seaworthiness of submarines. If they are able to operate well below the turbulent surface, they will be insulated from wind and wave action and will suffer no degradation of speed because of adverse conditions on the surface.

Relative Speed Degradation of Alternative Platforms

The relative seaworthiness of the various platform concepts, as based on the qualitative discussions above, is shown conceptually in Fig. 32.

Planing hulls, as shown, presently exhibit very poor qualities. For large planing hulls, immediate loss of speed is expected as sea conditions worsen. The characteristics of the GEM are shown as a band, with the bottom portion representing present vehicles without skirts and the top indicating possible performance improvement with skirts or the performance capability of the largest GEM's. Testing of ocean-going GEM's is critical to obtaining a full assessment of the seakeeping characteristics of this platform concept. The displacement hull curve represents the general characteristics exhibited by all displacement hulls. Clearly, there is some variation on either side of this curve, with the performance of any given vessel dependent on the exact shape of the hull employed and the actual speed characteristics. If the work at DTMB proves successful, high-speed displacement hulls could experience a substantial improvement in rough-sea capabilities. The hydrofoil is expected to exhibit a high degree of seaworthiness, as shown. Large hydrofoils would probably have better adverse-sea capabilities than any concept other than the submarine. The submarine has excellent seakeeping qualities, largely due to its ability to operate beneath the turbulent surface. As the figure indicates, submersibles may be considered insensitive to surface conditions except in close waters.

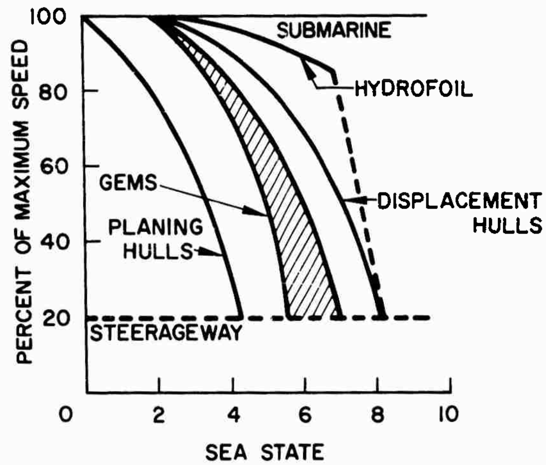


FIG. 32 GENERAL CAPABILITIES OF VARIOUS PLATFORM CONCEPTS
SPEED DEGRADATION IN HEAD SEA

TECHNICAL NOTES

The following brief technical notes, with accompanying references, provide background information and discussion on the potential for reducing the resistance of displacement hulls, on the use of weight-saving materials in shipbuilding, and on alternative power and propulsion systems. These notes were prepared by M. Rosenblatt & Son, Inc., and are included in the study to provide a clearer appreciation of critical technical areas in consideration of alternative platform concepts.

I. Methods for Reducing Resistance of Displacement Hulls

The total resistance experienced by a ship in a seaway consists of:

Still-water resistance which, based on Froude's assumption, may be further broken down into two parts: (a) frictional resistance and (b) residual resistance (mainly due to wave-making).

Increased resistance due to irregular waves.

A ship's speed may be increased by reducing resistance. This does not necessarily mean, however, that a ship can operate at high speeds in rough seas just because enough thrust has been provided to overcome the resistance. In fact, the seaworthiness of a ship (that is, ship motions, increased resistance, and other seagoing qualities), which is a function of the hull form, Froude number, and the surrounding seas, is a major factor which controls the obtainable speed. Hence, in designing a practical hull form of less total resistance or wave-making resistance, the seaworthiness of the form must be considered.

Some proposed methods of reducing frictional resistance, and some practical hull forms of less wave-making resistance will be briefly discussed, with due consideration of seaworthiness.

Devices To Reduce Frictional Resistance

The following two methods of reducing frictional resistance are those discussed in Reference (1).[1]

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1. References are given at end of each note.

(1) Sucking off of the boundary layer at various places along the length of the body. In addition to the difficulties raised in Reference (1), it is doubtful that the gain in power due to the reduction of frictional resistance would be large enough to offset the extra power required to provide the suction.

(2) Use of a coating that absorbs the energy in perturbations in the water. The theory and an experiment, conducted primarily for the purpose of reducing the frictional drag of under-water missiles, are described in Reference (6). To date, no significant drag reduction has been obtained.

Neither of these two methods nor any others proposed seem to hold out much promise for significant reductions in frictional resistance for the design of vessels to be operational in the 1975-1980 time period.

Hull Forms of Less Wave-Making Resistance

Wave-making resistance is that part of the total resistance experienced by a ship in still water that can be expected to be improved by changes in hull form.

Much work has been done to obtain an understanding of wave-making resistance and to develop hull forms of least wave-making resistance. Brief discussions of some practical hull forms of possible use in an amphibious fleet follow.

Bulbous Bow

Reduction in wave-making resistance with the use of a bulbous bow has been known for many years. Bulbous bows of cross-sectional area equal to approximately 10 percent of midship section area have been adopted by many seagoing vessels. These bulbs reduce the effective horsepower requirement for a given speed by approximately 8 percent. As to the seaworthiness of the ships with these bulbous bows, however, Reference (5) states "A wide variation in bulb sizes has been found to have less effect than originally expected with regard to resistance, motions, and other quantities measured in head seas."

Recently, Inui has developed a new bulb theory (Reference (3)) using extremely large sizes of bulb. A few studies on the practical application of Inui's theory have been published (see Appendix 3 of Reference (3)).

Reference (4) deals with the resistance in still water and the performance in waves of the conventional high-speed escort destroyer (C_p = 0.62, Froude no. = 0.5 or $V/\sqrt{L}$ = 1.7, L = 383 feet, and model length = 26'-9") with a large bulbous bow. The author drew the following conclusions:

1. The shape and location of the bulb have a great influence on the effectiveness of the bulb.

2. For most acceptable designs (spherical bulbs of approximately 25 percent of midship cross section), the reduction in residual resistance from that of the parent models is about 30 percent in the speed range of 18 to 20 knots.

3. The increase in frictional resistance due to the increase in wetted surfaces of the bulb cancels part of the gain in residual resistance. The estimated net reduction in effective horsepower acquired is in the order of 7 percent in the speed range of 18-20 knots.

4. The large bulbous bow decreases the ship motion, and its effect is most marked at high speeds and in short waves.

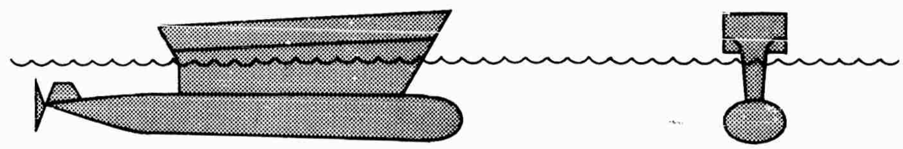
5. The thrust requirement due to waves is slightly greater with bulb than without bulb.

In view of the above it may be stated that with proper design of the bulbous bow, either conventional size (approximately 10 percent) or large size (approximately 25 percent), it is possible to reduce EHP requirements approximately 7 percent or 8 percent within the state-of-the-art.

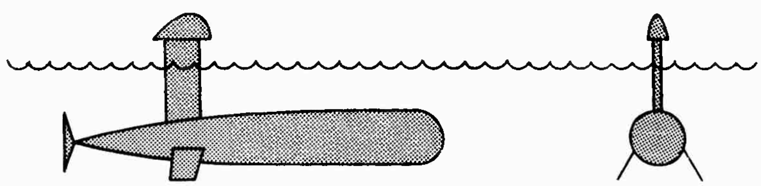
Improvements of Existing Hull Forms

Mathematical descriptions of hull forms of least wave-making resistance or total resistance have been developed by a number of people. Although these forms are impractical, they are of great importance in

suggesting ways of improving the existing hull forms. One approach to greatly reducing wave resistance is to place the main body of the hull below the surface, as illustrated below:



Shark Form



Semi-Submarine (Near-Surface Craft)

These designs are based on the principle of avoiding the free-surface effect (wave resistance) by submerging the main hull below the water surface. At low Froude number, the total resistance of these craft is greater than that of a destroyer of the same size, due to the greater frictional resistance resulting from the greater wetted surface and the relative unimportance of wave resistance for the surface ship at low Froude number. However, at high Froude number these craft are better than destroyer forms (see Figs. 6 and 14 of Reference (1)).

The shark-form craft suffers severe motions in stern seas (Reference (2)). The near-surface craft has exceptionally long pitching and heaving periods which permit it to operate at any speed in head seas of any severity and at high speeds in stern seas of moderate severity (Reference (2)). Motion control for these ship types, however, can probably be achieved with small dynamic control surfaces. There is only limited topside space in the shark form and none in the near-surface craft. Hence, the nature of the mission will decide the applicability of these craft in addition to resistance considerations and seaworthiness.

Reference (2) contains a much more complete evaluation of novel ship types than can be made within the time limits of this study. However, a few of the major conclusions of Reference (2) are quoted below:

1. "The existing destroyer type of ship has advantages over any of the proposed new ship types in terms of its ability to carry a larger payload weight at a specified speed and endurance in moderate weather."

2. "Application of the new technology in the field of power plants, that is being utilized in the current hydrofoil programs, to the design of surface ship types would also permit significant advances in their performance . . . . Furthermore, contrary to much current opinion, the range of sea states in which maximum speed could still be sustained would also be considerably enhanced by this increase in power concentration."

References

1. Oakley, O.H., High Performance Ships--Promises and Problems, Third Symposium on Naval Hydrodynamics, Scheveningen, Netherlands, September 1960.

2. Mandel, P., A Comparative Evaluation of Novel Ship Types, Transactions SNAME, Vol. 70, 1962.

3. Inui, T., Wave-Making Resistance of Ships, Transactions SNAME, Vol. 70, 1962.

4. Takejawa, S., A Study on the Large Bulbous Bow of a High-Speed Displacement Ship, Part 1 - Resistance Tests in Still Water, Part II - Performances in Waves, translated by Dr. Takahei, University of Michigan, 1961.

5. Dillon, E.S. and E.V. Lewis, Ships with Bulbous Bows in Smooth Water and in Waves, Transactions SNAME, Vol. 63, 1955.

6. Boggs, F.W., H.R. Frey, and E.B. Hahn, Construction and Testing of Coating for Drag Reduction, United States Rubber Company, February 1, 1962.

II. Lightweight Materials

In this discussion the potential for weight saving in ship construction through the use of either high strength steels (compared with mild steel), or aluminum is investigated. It is clear that some weight could be saved by judicious use of high-strength steels in hull frames, deck plates, foundations, and the like. However, appreciable weight savings would result only from the use of a lighter-weight material with strength equivalent to that of mild steel, such as aluminum. Presently, the destroyer-type ship combines high-strength steels for the hull with aluminum for high (deckhouse) areas. Use of aluminum for the hull should result in weight savings up to 30-60 percent of the present hull weights, depending on how much deflection is permitted. Complete weight studies for various-type aluminum hull ships should be undertaken. The characteristics of high-strength steel construction and aluminum construction are considered below.

High-Strength Steels

The higher-strength steels have the same elastic properties as mild steel, but yield points may vary up to a presently workable construction grade steel yield point of 100,000 psi. These higher-strength steels have superior corrosion-resisting properties (compared with mild steel) but are formed and welded with somewhat greater difficulty (and expense). Weight savings can result if the plate thickness can be reduced.

Where plating sections are stressed in edgewise compression, buckling becomes the criterion for strength, and very high yield points do not enable thicknesses to be reduced except where the plating is relatively thick (about one inch or over). Generally, naval ship hull plating runs below one inch. Strength deck plates might run as high as $1\frac{1}{2}$ inches in midship areas, and some savings may be realized here.

Tables IV and V show the effect of buckling action on various-size plating panels. The width/thickness ratio determines whether a panel acts as a short, intermediate, or long column. The short column permits the yield point to be used for comparison, and in this range higher strength materials markedly affect weight-savings potential. Intermediate range columns reduce the strength somewhat, reducing the effect of higher strength steels. Long columns result in no weight savings through use of higher strength materials.

Table VI shows comparative thickness for plating panels loaded on edges. This is the situation for elements of the hull girder in bending and applies to deck and shell plating.

The following will explain the abbreviations used in Tables IV, V, and VI:

b = panel width; 30" is typical for most ships

T = plating thickness

F_c = buckling strength ksi

HT = steel with YP = 50 ksi

HY = steel with YP = 99 ksi

Table IV

STRENGTH OF STEEL PLATING PANELS IN EDGE COMPRESSION

		Mild Steel		HT		HY	
T	b = 30" b/T	Column Range	F_c	Column Range	F_c	Column Range	F_c
1-1/2	20	Short	33	Short	50	Short	99
1-1/4	24	Short	33	Short	50	Short	99
1	30	Short	33	Short	50	Inter.	80
7/8	34	Short	33	Short	50	Inter.	67
3/4	40	Short	33	Inter.	43	Long	49
5/8	47	Inter.	30	Inter.	35	Long	35
1/2	60	Inter.	22	Inter.	22	Long	22

Source: Reference (1).

Table V

STRENGTH OF STEEL PLATING PANELS WITH ADDED LONGITUDINALS TO CUT DOWN PANEL WIDTH

	b = 24"	Mild Steel		HT		HY	
T	b/T	Column Range	F_c	Column Range	F_c	Column Range	F_c
1-1/2	16	Short	33	Short	50	Short	99
1-1/4	19.2	Short	33	Short	50	Short	99
1	24	Short	33	Short	50	Short	99
7/8	27.4	Short	33	Short	50	Inter.	88
3/4	32	Short	33	Short	50	Inter.	73
5/8	37.4	Short	33	Inter.	46	Long	56
1/2	48	Inter.	29	Inter.	34	Long	34

Source: Reference (1).

Table VI

THICKNESS REDUCTION IN STEEL PLATING PANELS FOR EQUIVALENT STRENGTH

Mild Steel	b = 30"		b = 24"	
Panel b = 30"	HT	HY	HT	HY
1-1/8	7/8	*	7/8	3/4
1	7/8	*	3/4	*
7/8	*	*	3/4	*

Note: Thickness includes corrosion allowance of 1/8.

*Thickness cannot be decreased by increasing strength. Long column buckling is the criterion here.

Source: Reference (1).

Use of the high-strength steels for main hull girder elements does not result in appreciable weight savings (calculated values of savings for AKA-112 are under 25 tons). Using a material with higher yield point than 50,000 psi does not result in any weight savings (compared with the 50,000 psi material).

The high-strength steels should lend themselves to specialized uses, such as massive foundations, struts, hatch girders, and the like, although even here there is a lower limit to which weight can be reduced, below which vibration characteristics rather than strength become the primary consideration.

Data are not readily available on weight-savings figures for optimized vessels of high-strength steels, although the Navy destroyer types make full use of higher-strength steels and lightweight materials.

Aluminum

Potentially, the greatest weight savings could be realized through the use of aluminum. No large ship has been fabricated fully of aluminum, but several smaller craft are in use by the Navy and many ships have extensive aluminum components. Destroyer types have aluminum deck houses. The missile cruisers have been completely stripped above the main deck, and deck houses of aluminum have been installed to increase the payload capacity for missile handling. Other applications have been masts, stacks, hatch covers, gratings, and platforms.

Aluminum is superior to steel in corrosion resistance but, since it has one-third the modulus of elasticity, structural members deflect three times the amount of equal steel members. Yield stresses compare favorably with mild steel, and future developments may result in higher strength alloys. Material costs of aluminum run 6-8 times those of mild steel. A representative cost comparison is given in Table VII.

References

1. Priest and Gilligan, Design Manual for High Strength Steels.

2. C. H. Holtyn, Aluminum From Boats to Ships, Reynolds Metal Company.

Table VII

APPROXIMATE COMPARATIVE COSTS OF STEEL AND ALUMINUM
FOR SMALL TANKERS

	130-Foot London Design		219-Foot German "Aluminia"		222-Foot Ocean Tanker	
	Steel	Aluminum	Steel	Aluminum	Steel	Aluminum
LBP	130'	130'	219'-2"	219'-2"	222'	222'
Beam	25'-6"	25'-6"	26'-8"	26'-8"	37'	37'
Depth	9'	9'	12'-1"	12'-1"	19'-3"	19'-3"
Oper.Draft	8'	8'	8'	8'	17'	17'
Hull Wt.	127 tons	54.2 tons	241 tons	91 tons	720 tons	310 tons
Machy. Wt.	9.7	9.7	67	67	130	130
Outfit Wt.	23.3	18.7	26	26	170	170
Light Ship	160	82.6	334	184	1020	610
Total DWT	505	582	800	950	2010	2420
DWT Gained		77.4 (15.3%)		150 (18.8%)		410 (20.3%)
Total Cost	$174,900	$216,000	$419,500	$519,000	$1,101,000	$1,497,000
Cost/ton DWT	$346	$372	$524	$546	$548	$617

Source: Reference (2).

III. Power and Propulsion Systems

Power Plants

Many types of propulsion plants have been investigated as to the present state-of-the-art and as to possible future developments that may be available in the 1975 to 1980 time period.

The following is a list of the types of propulsion plants consioered:

1. Geared steam turbine

2. Diesel propulsion

   a. Low speed, direct drive
   b. Medium speed, geared
   c. High speed, lightweight

3. Industrial-type gas turbine

4. Aircraft-type gas turbine

5. Nuclear powered

6. Fuel cells.

The geared steam turbine presently powers the majority of surface craft in the U.S. Navy amphibious fleet. This type of main propulsion system has a very low weight per SHP ratio for horsepowers greater than 50,000. In the lower SHP range, below 50,000, there is a sharp increase in the WT/SHP ratio, as shown graphically in Fig. 33. For the present state-of-the-art, the lowest WT/SHP is approximately 20 lbs/SHP.

From reports on recent and future developments, this ratio may be cut by approximately 15 percent using an integrated steam plant. This type of plant will also reduce the space requirements. Present and projected fuel rates are, respectively, 0.507 lb/SHP/HR and 0.47 lb/SHP/HR (see Reference (6)).

Diesel propulsion may be divided into the following categories:

Low speed, direct drive, 150 rpm and under
Medium speed, geared, 150 rpm to 750 rpm
High speed, lightweight, 750 rpm and over

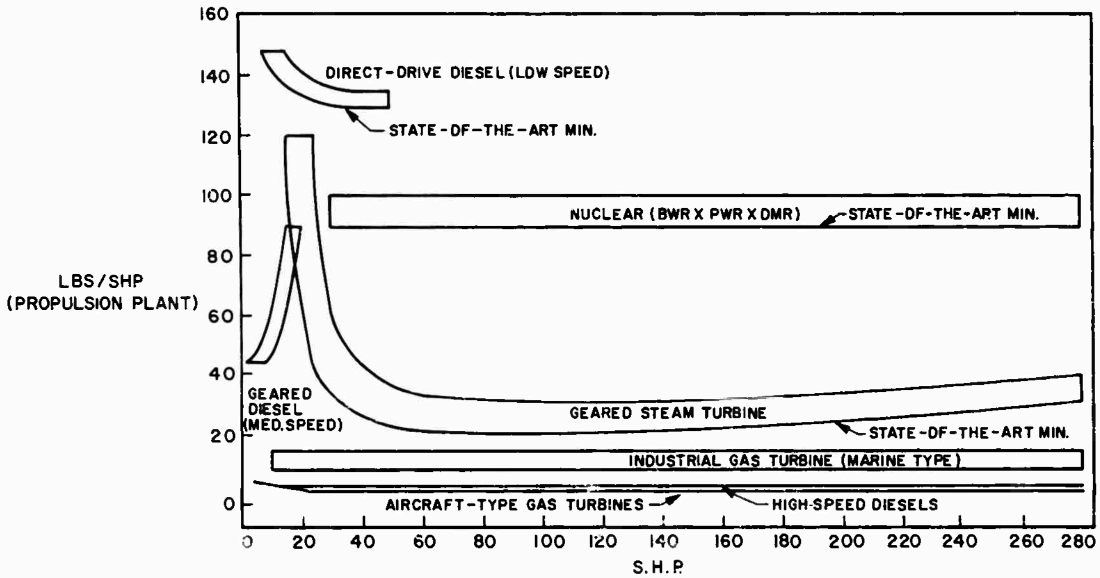


FIG. 33 PROPULSION PLANT STUDY

Low-speed, direct-drive diesels are heavy, 130 lbs/SHP (see Fig. 33), and occupy a great deal of space. The present maximum power attainable from one engine is 30,000 SHP. The fuel rate is rather low, approximately 0.32 lb/SHP/HR. Future developments in this type of engine will continue at a slow pace with little improvement over the present WT/SHP ratio or fuel rate.

Medium-speed, geared diesels provide versatility, and units may be compounded through gearing and clutches to provide the desired SHP or used in conjunction with other types of power plants, such as gas turbines. Weight/SHP ratios for geared diesels for the present state-of-the-art are approximately 40 lbs/SHP (see Fig. 33). The fuel rate is slightly higher than for low speed diesels, approximately 0.34 lb/SHP/HR. Many refinements have been made and, with the use of improved techniques and materials, the weight/power ratio may be further reduced.

High-speed diesels with a very low weight/power ratio, approximately 4.0 lbs/SHP (see Fig. 33), have all the advantages of the medium-speed diesel, plus the advantage that they may be used on installations where space requirements are limited. Fuel rate is approximately 0.4 lb/SHP/HR.

Industrial-type gas turbines using gasifiers and necessary auxiliary equipment have weight/power ratios in the range of 9 lbs/SHP to 15 lbs/SHP for the present state-of-the-art. Fuel rate is approximately 0.52 lb/SHP/HR. Future developments of this type of power plant using more extensive welded fabrication and improved materials indicate that a decrease in the weight/power ratio and fuel rate will be forthcoming.

The aircraft gas turbine shows great promise in the field of high-speed surface craft used either as a prime mover or as part of a combined plant. The weight/power ratio is the lowest of all the plants investigated, 2.5 lbs/SHP, with a fuel rate of 0.55 lb/SHP/HR. This power plant is most efficient at maximum RPM, but has very poor idling characteristics. Very high SHP limits can be attained by compounding several units of this type.

Marine-type nuclear reactor propulsion plants presently installed on surface and underwater vessels fall into two categories, the boiling water reactor type (BWR) and the pressurized water reactor type (PWR), both with steam turbine machinery. The weight/power ratio for these propulsion systems for the state-of-the-art is approximately 90 lbs/SHP (see Fig. 33). The organic moderated reactor is still in the development stage; however, its specific weight seems to be about the same as that of the BWR and PWR (see references (2) and (5)). Recent developments in gas-cooled reactors, based on developments arising out of the aircraft nuclear power program,

show great promise, with possible weight/power ratios of from 7 to 45 lbs/SHP. The specific weight decreases as the amount of horsepower increases and with the substitution of gas turbines for steam turbines (see References (4) and (7)).

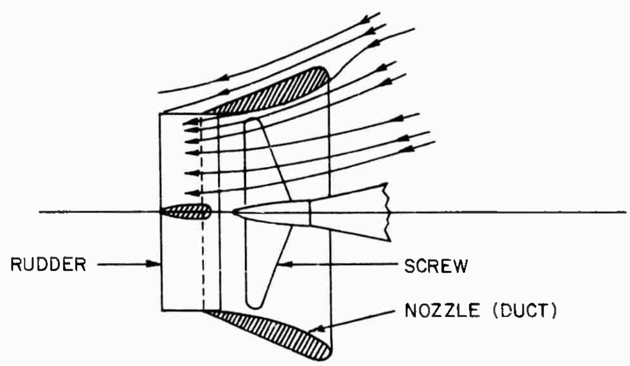
Propulsion plants using <u>fuel cells</u> as a power source are still in the very preliminary research and experimental stages. Such research indicates that in the near future an operating power plant may be available with a weight/power ratio of approximately 45 lbs/SHP. The potential advantage of the fuel cell is its high over-all efficiency.

Propulsion Systems

Supercavitating Propeller

The operation of conventional propellers is sometimes associated with the problem of cavitation. When cavitation occurs and becomes serious, the efficiency of the propeller is greatly reduced. In addition, the collapse of the vapor bubbles is one of the causes of blade surface erosion, noise, and vibration. When the installation of a conventional propeller would unavoidably lead to severe cavitation for ships with speeds above about 35 knots, for example, a supercavitating propeller may be used. A conventional (subcavitating) propeller of good design operating without cavitation is always more efficient than a supercavitating propeller. However, in the region where serious cavitation is unavoidable, the supercavitating propeller is more efficient and eliminates the problem of damage to the blade surfaces and other deficiencies associated with cavitation.

Ducted Propeller ("Nozzle and Screw" Propeller)



As shown in the illustration, this thrust device consists of a propeller operating inside a circular duct or nozzle having an airfoil cross section. Systematic experiments (references (8) and (9)) show that a well-proportioned nozzle-propeller combination for large screw loadings[1] is always superior to the propeller alone in efficiency, thrust developed, and performance in waves and also reduces the chance of propeller breakage and the like.

While the nozzle-propeller combination has not been developed sufficiently, further research may prove it to be a potential alternative to the supercavitating propeller for the thrusting device of high-speed ships.

Water Jet Propulsion

The thrust force of this device is obtained by means of ejecting water through underwater nozzle(s). For crafts of large gross weight and low speed, such as ships (even high-speed ships are comparatively slow), a jet propulsion device of any sort has poor efficiency. Hence, water jet propulsion is not considered suitable for large ships.

References

1. Marine Gas Turbines, Pratt & Whitney Aircraft

2. Economics of Nuclear and Conventional Merchant Ships, June 1958

3. Study of Hydrofoil Seacraft, Grumman Aircraft Engineering Corporation, October 1958 (Vol. I and II only)

4. "Nucleonics," June 1963

5. Krasnos, A.W., Nuclear Propulsion for Merchant Ships, 1962

6. MacMillan, D.C., Improved Steam Propulsion Plant, SNAME paper, November 16, 1962

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1. An increase in efficiency with the use of a nozzle can be obtained even at very low screw loading (Reference (9))--$B_p = \frac{N\ P^{0.5}}{V_a^{2.5}} = 1.3$, where N = rpm, P = SHP, V_a = advance velocity in knots.

7. "Maritime Nuclear Steam Generator," Status Report No. 1, General Electric Company, September 12, 1963

8. Solovev, V. I. and D. A. Chumak, Experimental Investigation of Propellers in Nozzles at the Central Aero-Hydrodynamic Institute (Moscow), DTMB Translation 302, March 1961

9. Van Manen, T. D., Recent Research on Propellers in Nozzles, Journal of Ship Research, July 1957

IV. Description of Sea States

When the operational capabilities of alternative platform concepts are discussed, such phrases are used as "ability to operate in sea state 3," or "must survive in sea state 6." It is necessary, therefore, to define what is meant by these references and to indicate how often adverse sea conditions are likely to occur.

Table VIII represents the most complete description of sea states that the project team was able to locate. A brief discussion of the various sea states is included below, and should be read in conjunction with the table.

Sea state 0. Sea like a mirror; flat calm; very little wind. It is probable that sea state 0 will never be encountered in the open sea. This is because there are always some old or decayed swells present in the open sea, which imperceptibly disturb the surface of the ocean.

Sea states 1, 2, and 3. These conditions cover the vast majority of situations encountered in rivers, harbors, and protected bays and inlets. Sea state 3 is normally considered to be the point of separation between a slight or choppy sea and a more developed deep-water disturbance. The initial [illegible] hydrofoil test vehicles were designed to operate in a state 3 [illegible]

Sea states 4, 5, and 6. These conditions of wind and sea cover the vast majority of "adverse sea conditions" encountered in the open ocean. The maximum wave height reported in any part of the ocean by U.S. Naval officers during the three years 1883-86 was 25 feet see Reference (1), p. 23). Similarly, the largest waves encountered by the H.M.S. Challenger during a scientific cruise around the world in 1873-75 were only 18-22 feet high (see again Reference (1), p. 23. It will be seen in Table VIII that these wave heights correspond to an advanced state 6 sea.

Sea states 7, 8, and 9. These sea states represent the gale, whole gale, and hurricane conditions in the oceans. Most authorities agree that there has never been a fully developed hurricane sea (see Reference (1), p. 74). This is because a "fetch" of over 2,500 nautical miles is required to produce a hurricane sea. "Fetch" refers to a wind of a given force, blowing over open water, in a constant direction for the required stated distance. Since a hurricane invariably follows

Table VIII

DESCRIPTION OF SEA STATES

WIND AND SEA SCALE FOR FULLY ARISEN SEA														
SEA-GENERAL		WIND[3]				SEA[3]								
						WAVE HEIGHT FEET								
SEA STATE[1]	DESCRIPTION[2]	WIND FORCE (BEAUFORT)	DESCRIPTION	RANGE (KNOTS)	WIND VELOCITY (KNOTS)	AVERAGE	SIGNIFICANT	AVERAGE 1/10 HIGHEST	SIGNIFICANT RANGE OF PERIODS (SECONDS)	T_{max} (PERIOD OF MAXIMUM ENERGY OF SPECTRUM)	$\tilde{T}$ (AVERAGE PERIOD)	$\mathcal{L}$ (AVERAGE WAVE LENGTH)	MINIMUM FETCH (NAUTICAL MILES)	MAXIMUM DURATION (HOURS)
0	Sea like a mirror.	0	Calm	Less than 1	a) 0	0	0	0	–	–	–	–	–	–
	Ripples with the appearance of scales are formed, but without foam crests.	1	Light Airs	1 – 3	2	0.05	0.08	0.10	Up to 1.2 sec.	0.7	0.5	10 in.	5	18 min.
1	Small wavelets, still short but more pronounced; crests have a glossy appearance, but do not break.	2	Light Breeze	4 – 6	5	0.18	0.29	0.37	0.4 – 2.8	2.0	1.4	6.7 ft.	8	39 min.
	Large wavelets, crests begin to break. Foam of glossy appearance. Perhaps scattered white horses.	3	Gentle Breeze	7 – 10	$\boxed{8.5}$	0.6	1.0	1.2	0.8 – 5.0	3.4	2.4	20	9.8	1.7 hrs.
					10	0.88	1.4	1.8	1.0 – 6.0	4	2.9	27	10	2.4
2	Small waves, becoming larger, tairly frequent white horses.	4	Moderate Breeze	11–16	12	1.4	2.2	2.8	1.0 – 7.0	4.8	3.4	40	18	3.8
					$\boxed{13.5}$	1.8	2.9	3.7	1.4 – 7.6	5.4	3.9	52	24	4.8
3					14	2.0	3.3	4.2	1.5 – 7.8	5.6	4.0	59	28	5.2
					16	2.9	4.6	5.8	2.0 – 8.8	6.5	4.6	71	40	6.6
4	Moderate waves, taking a more pronounced long form; many white horses are formed (chance of some spray).	5	Fresh Breeze	17–21	18	3.8	6.1	7.8	2.5 – 10.0	7.2	5.1	90	55	8.3
					$\boxed{19}$	4.3	6.9	8.7	2.8 – 10.6	7.7	5.4	99	65	9.2
					20	5.0	8.0	10	3.0 – 11.1	8.1	5.7	111	75	10
5	Large waves begin to form; the white foam crests are more extensive everywhere (probably some spray).	6	Strong Breeze	22–27	22	6.4	10	13	3.4 – 12.2	8.9	6.3	134	100	12
					24	7.9	12	16	3.7 – 13.5	9.7	6.8	160	130	14
					$\boxed{24.5}$	8.2	13	17	3.8 – 13.6	9.9	7.0	164	140	15
6					26	9.6	15	20	4.0 – 14.5	10.5	7.4	188	180	17
	Sea heaps up and white foam from breaking waves begins to be blown in streaks along the direction of the wind (spindrift begins to be seen).	7	Moderate Gale	28–33	28	11	18	23	4.5 – 15.5	11.3	7.9	212	230	20
					30	14	22	28	4.7 – 16.7	12.1	8.6	250	280	23
					$\boxed{30.5}$	14	23	29	4.8 – 17.0	12.4	8.7	258	290	24
					32	16	26	33	5.0 – 17.5	12.9	9.1	285	340	27
7	Moderately high waves of greater length, edges of crests break into spindrift. The foam is blown in well-marked streaks along the direction of the wind. Spray affects visibility.	8	Fresh Gale	34–40	34	19	30	38	5.5 – 18.5	13.6	9.7	322	420	30
					36	21	35	44	5.8 – 19.7	14.5	10.3	363	500	34
					$\boxed{37}$	23	37	46.7	6. – 20.5	14.9	10.5	376	530	37
					38	25	40	50	6.2 – 20.8	15.4	10.7	392	600	38
					40	28	45	58	6.5 – 21.7	16.1	11.4	444	710	42
8	High waves. Dense streaks of foam along the direction of the wind. Sea begins to roll. Visibility affected.	9	Strong Gale	41–47	42	31	50	64	7. – 23	17.0	12.0	492	830	47
					$\boxed{44}$	36	58	73	7. – 24.2	17.7	12.5	534	960	52
					46	40	64	81	7. – 25.	18.6	13.1	590	1110	57
	Very high waves with long overhanging crests. The resulting foam is in great patches and is blown in dense white streaks along the direction of the wind. On the whole the surface of the sea takes a white appearance. The rolling of the sea becomes heavy and shocklike. Visibility is affected.	10	Whole Gale*	48–55	48	44	71	90	7.5 – 26	19.4	13.8	650	1250	63
					50	49	78	99	7.5 – 27.	20.2	14.3	700	1420	69
					$\boxed{51.5}$	52	83	106	8. – 28.2	20.8	14.7	736	1560	73
9					52	54	87	110	8. – 28.5	21.0	14.8	750	1610	75
					54	59	95	121	8. – 29.5	21.8	15.4	810	1800	81
	Exceptionally high waves (small- and medium-sized ships might for a long time be lost to view behind the waves). The sea is completely covered with long white patches of foam lying along the direction of the wind. Everywhere edges of wave crests blown into froth. Visibility affected.	11	Storm*	56–63	56	64	103	130	8.5 – 31.	22.6	16.3	910	2100	83
					$\boxed{59.5}$	73	116	148	10. – 32.	24.	17.0	985	2500	101
	Air filled with foam and spray. Sea completely white with driving spray. Visibility very seriously affected.	12	Hurricane*	64–71	>64	>80 b)	>128 b)	>164 b)	10. – (35)	(26)	(18)	–	–	–

* For hurricane winds (and often whole gale and storm winds) required durations and fetches are rarely attained. Seas are therefore not fully arisen.

a) A heavy box around this value means that the values tabulated are at the center of the Beaufort range.

b) For such high winds, the seas are confused. The wave crests blow off and the water and air mix.

1) Encyclopedia of Nautical Knowledge, W. A. McEwan and A. H. Lewis, Cornell Maritime Press, Cambridge, Maryland, 1953, p. 483.

2) Manual of Seamanship, Vol II, Admiralty, London, H. M. Stationery Office, 1952, pp. 717-718.

3) Practical Methods for observing and Forecasting Ocean Waves, Pierson, Neuman, James., New York University College of Engineering, 1953.

Source: Table compiled by Wilton Marks, David Taylor Model Basin.

either an irregular or a curved track in the ocean, there has never been a true hurricane sea. In fact, during the hurricane month of August in the Caribbean for the years 1887 to 1936, only 51 storm seas (state 7) were recorded, or an average of about one per year[2] (see again Reference (1), p. 74).

The highest wave height ever reliably reported was 112 feet, encountered by the U.S.S. Ramapo in the North Pacific on November 7, 1933 (see Reference (1), pp. 23-24). This was an extremely rare instance. In fact, waves much higher than 25 feet are not usual anywhere on the oceans. It would appear that any vehicle capable of operating in an advanced state 6 sea would essentially fulfill the seaworthiness limits expected to be encountered in her lifetime.

With reference to frequency of occurrence, information has been compiled on the basis of 40,164 log-book entries of sailing ships (see Reference (2), p. 21), for the years 1925-1928. The compilation of this information is shown in Table IX.

The recorded information available indicates that a vehicle designed to operate in sea states 5 to 6 will be able to perform the vast majority of assigned missions satisfactorily.

References

1. Bigelow, H.B. and W.T. Edmonson, Wind Waves at Sea, Breakers and Surf, H.O. No. 602.

2. Schumacher, Arnold, 1939 Stereophotogrammetrische Wellenaufnahmen, Wissenschaftliche, Ergebnisse der Deutschen Atlantischen Expedition auf dem Forschungsund Vermissungschiff "meteor" 1925-28, as reported in Wind Waves at Sea, Breakers and Surf.

Table IX

RELATIVE FREQUENCY OF SEA STATES IN DIFFERENT REGIONS

Area	Sea State (percents)					
	0-2	3	4	5	6	>6[a]
N. Atlantic - Newfoundland to England	20	20	20	15	10	15
Mid-equatorial Atlantic	20	30	25	15	5	5
North Pacific - Oregon to Alaska	25	20	20	15	10	10
East equatorial Pacific	25	35	25	5	5	5
North Indian Ocean						
Northeast monsoon season	55	25	15	5	0	0
Southwest monsoon season	15	15	25	20	15	10
South Indian Ocean	30	25	20	15	5	5

a. Some advanced state 6 seas included in this figure.

Source: Reference (2), Table 8, p. 21.

Appendix A

CONTENTS

ILLUSTRATIONS

Illustrations (continued)

Illustrations (continued)

Illustrations (continued)

Illustrations (continued)

Illustrations (concluded)

Appendix A

ADVANCED DISPLACEMENT HULLS

Appendix A

ADVANCED DISPLACEMENT HULLS

Introduction

In this appendix the major characteristics of high-speed displacement ships for possible use in the amphibious fleet of 1970-1980 are analyzed and presented. Speeds to 70 knots, displacements to 40,000 tons, and ranges to 3,000 nautical miles are considered and the corresponding payloads and shaft horsepowers determined.

Resistance and Powering

Hull Form

The hull form of Referencc No. 1, with displacement-length ratio of sixty (60) and beam-draft ratio of 2.25, was used for estimating resistance. This form has superior high-speed residual resistance characteristics, and is compatible with strength and stability requirements. The smooth-water residual resistance of this hull form, at speed-length ratios above 1.2, is approximately 10 percent better than that of Taylor's Standard Series.[2] The corresponding decrease in total resistance is approximately 6 percent.

Resistance

The smooth-water resistance was obtained by expanding the model test data of Reference No. 1 to full size. Residual and frictional resistance were expanded, respectively, in accordance with the laws of hydrodynamic similitude[3] and the Schoenherr mean line.

A roughness allowance of 0.0004 has been assumed to account for the clean surface roughness of the full size ships.[4] This is in accordancc with accepted practice.

Horsepower Requirements

Effective horsepower (EHP) was calculated by the relation: EHP $\frac{R(v)}{550}$ where: R = ship's resistance (smooth water) in lbs and v = ship's speed in ft/sec.

In the shaft horsepower (SHP) calculations, the propulsive coefficient (PC) was assumed to be 0.65. A thirty percent (30%) service horsepower margin was also made. Shaft horsepower was then calculated by the relation: $SHP = \frac{EHP}{0.65} \times 1.30 = \frac{EHP}{0.50} = 2EHP$.

Curves of SHP plotted as a function of speed, for a number of displacements, are shown in Figs. A-1 and A-2. Curves of SHP/ton plotted as a function of speed, for the same displacements, are shown in Fig. A-3.

Propulsion System

A propulsion system consisting of geared steam turbines and water propellers has been assumed.

Weights

General

A ship's weight is defined by the following equation:

$$\Delta = \Delta_L + D.W.$$

$$\Delta = W_H + W_{PM} + W_A + W_O + W_F + W_{PL}$$

where:

Δ = full load displacement

Δ_L = $W_H + W_{PM} + W_A + W_O$ = light ship displacement

D.W. = $W_F + W_{PL}$ = deadweight

W_H = hull structure

W_{PM} = propulsion system

W_A = auxiliary systems

W_O = outfit

W_F = fuel oil

W_{PL} = payload (includes crew).

The light ship weight components are related to BuShips standard light ship weight groups as follows:

W_H = BuShips Weight Group No. 1

W_{PM} = BuShips Weight Group No. 2

W_A = BuShips Weight Group Nos. 3 and 5

W_O = one-half of BuShips Weight Group No. 4 plus Weight Group No. 6.

BuShips Weight Group No. 7 and one-half of Group No. 4 are considered part of specific mission requirements and are therefore classified as part of payload. BuShips Weight Group classifications is indicated below:

BuShips Weight Group No.	Title
1	Hull structure
2	Propulsion
3	Electric plant
4	Communication and control
5	Auxiliary system
6	Outfit and furnishings
7	Armament

Hull Structure Weight (W_H)

The basic displacement-payload-horsepower curves developed for the advanced displacement hulls assume mild steel to be the major structural material. Improvement in the potential for this platform concept, if constructed of aluminum hulls and powered by lightweight gas turbine propulsion systems, is considered at the end of this appendix.

A comparison between hull structure weights given in Reference Nos. 5 and 6 is presented in Fig. A-4. In addition, the hull structure weights of many existing naval vessels are plotted in the figure, and a "state of the art minimum (SAM)" curve is drawn. Agreement between the three curves of Fig. A-4 is reasonably good. These curves are reproduced in Fig. A-5 as percentages of full load displacement. The "SAM" curve of Figs. A-4 and A-5 has been used in this study.

Propulsion System Weight (W_{PM})

The state-of-the-art weight range for marine geared steam turbine propulsion systems has been examined. The minimum weight of such systems is seen to approach twenty pounds per shaft horsepower (20 lbs/SHP). In this study, the propulsion system has been assumed to weigh 20 lbs/SHP. Total propulsion system weight as a function of SHP has been plotted in Fig. A-6. Nuclear propulsion and lightweight gas turbine propulsion are considered at the end of this appendix.

Curves of propulsion system weight plotted as a function of speed, for a number of displacements, are given in Figs. A-7 and A-8. These curves were constructed from Figs. A-1, A-2, and A-6. The procedure was to enter Figs. A-1 or A-2 at a given speed and displacement and obtain the corresponding SHP. The propulsion system weight for this SHP was then obtained from Fig. A-6.

Weight of Auxiliaries (W_A)

The weight of auxiliaries of existing naval vessels is plotted, as a percentage of full load displacement, in Fig. A-9. This weight varies between 5 and 14 percent. A constant figure of 10 percent has been assumed for this study.

Outfit Weight (W_O)

The outfit weight given in Reference No. 5 is plotted as a percentage of full load displacement in Fig. A-10. The outfit weight of many existing naval vessels is similarly plotted, and a "SAM" curve is drawn. Figure A-11 shows the same curves plotted as actual outfit weights rather than percentages. The "SAM" curve of Figs. A-10 and A-11 has been used in this study.

Fuel Oil Weight (W_F)

The weight of fuel oil needed was calculated using Breguet's range equation:

$$W_F = \Delta(1 - \frac{1}{e^x})$$

where: $x = \frac{(SFC)\ (SHP)\ R}{2240\ V\ \Delta}$ and

where: Δ = full load displacement in long tons

W_F = fuel weight in long tons

SFC = specific fuel consumption in lbs/SHP/hr

SHP = shaft horsepower

R = range in nautical miles

V = ship's speed in knots.

A specific fuel consumption of 0.50 lb/SHP/hr has been assumed for this study.

Deadweight and Payload (D.W. and W_{PL})

Curves of deadweight plotted as a function of speed, for a number of displacements, are shown in Figs. A-12 and A-13. These curves were constructed from Figs. A-4, A-5, A-7, A-8, A-9, A-10, and A-11 using the equation:

$$D.W. = \Delta - \Delta_L = \Delta - W_H - W_{PM} - W_A - W_O \quad .$$

Curves of payload plotted as a function of speed, for a number of displacements, at a constant range of 500 nautical miles are shown in Figs. A-14 and A-15. These curves were constructed from the deadweight curves of Figs. A-12 and A-13 using the equation:

$$W_{PL} = D.W. - W_F \quad .$$

W_F was calculated as described above.

Similar curves for ranges of 1,000, 1,500, 2,000, and 3,000 nautical miles are plotted in Figs. A-16 through A-23.

Summary Curves for Mild Steel Hulls, Steam Turbine, Non-Nuclear Power Plant

Displacement and Payload versus Speed and Required Shaft Horsepower

Curves of displacement plotted as a function of speed, for various payloads and shaft horsepowers, for constant ranges of 500, 1,000, 1,500, 2,000, and 3,000 nautical miles are shown in Figs. A-24 through A-28, respectively. The curves were constructed from Figs. A-1, A-2, and A-14 through A-23. They assume mild steel hulls and geared steam turbine propulsion.

Payload Potential and Other Major Weight Components As Percentage of Full Load Displacement

The major weight components are plotted as a percentage of full load displacement for speeds of 25, 30, 40, 50, 55, and 60 knots in Figs. A-29 through A-34, respectively. Variation of payload with range is also indicated. These curves were constructed from Figs. A-4, A-5, and A-7 through A-13. Again, these curves assume mild steel hulls and geared steam turbine propulsion plants.

Payload-Range Curves

The variation of payload with range, at constant displacements and speeds, is shown in Figs. A-35 and A-36. These curves were constructed from Figs. A-12 through A-23.

Length, Beam, and Draft As a Function of Total Displacement

Curves of ships' lengths, beams, and drafts plotted as a function of displacement are shown in Fig. A-37.

The curves for beam-draft ratio of 2.25 correspond to ships geometrically similar to the model used for estimating resistance (see above). Curves for similar ships but with beam-draft ratio of 3.00 are also included. An increase in total resistance of approximately 2-1/2 percent accompanies this increase in beam-draft ratio. The greater beam might be required for certain ship configurations to ensure adequate stability.

Summary Curves for Mild Steel Hulls, Nuclear Power Plant

Lightweight nuclear propulsion for the advanced displacement hull, mild steel construction, has been considered. Nuclear power plant weight per shaft horsepower is shown in Fig. A-38. Figures A-39 and A-40 show payload potential and other major weight elements as a percentage of total displacement for various displacements and speeds.

Summary Curves for Aluminum Hulls, Gas Turbine and Nuclear Power Plants

To provide the information required to determine the effects on the payload capacity and performance potential of a displacement hull of using lightweight materials for structures and lightweight power plants and reducing outfit and auxiliary weights, a set of curves (Figs. A-41 through A-58) was prepared based on the following assumptions:

1. Aluminum alloy hulls. (The hull weight was assumed to be one-half that of a steel hull. See Fig. A-43.)

2. Gas turbine power plant of 5 lbs/HP (see Figs. A-44 through A-56). Lightweight nuclear power plant (see Figs. A-57 and A-58; see also Fig. A-38 for weight of nuclear plants).

3. Instead of 30 percent, a 15 percent power increase was allowed for added resistance in seaways (service margin). (See Figs. A-41 through A-42.)

4. Outfit weight (including auxiliaries) of 10 percent (taken from hydrofoil study).

The preliminary calculation based on these assumptions shows that the light ship weight is as low as 25 to 35 percent of the displacement. Although no investigation regarding the stability of the ship was made, due to lack of time allowed, it was noted that, for the fine hull form assumed in this study, the metacentric height, $\overline{GM}$, may become zero or negative at the light ship condition or at lightly loaded conditions. Hence, it was arbitrarily assumed that at no time should the displacement, less payload, be allowed to become less than 40 percent of the total displacement. To fulfill this requirement, permanent ballast should be added whenever the sum of light ship weight and fuel weight comes to less than 40 percent of the total displacement. Further study of the stability characteristics of the hull form is warranted. It is assumed that, as fuel is consumed, water ballast can be added to achieve stability.

A large increase in payload results from the reduction in other weight components, mainly hull weight and outfit weight. However, the following should be considered. A rough check shows that, except for dense cargoes (stowage factor of less than approximately 60 cubic feet/ton), a payload of more than 50 percent of displacement may not be obtained due to the volume limitations. A further investigation should be directed to ascertain the relationship between the maximum payload (weight) and stowage factor of cargoes for each particular hull form and power plant combination.

It was shown that the propulsive power plant weight is very small. Hence, it may be more desirable for practical designs to use a heavier plant with consequent reduction of cost, especially in the case when permanent ballast is required due to stability considerations.

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2. Gertler, Morton, A Reanalysis of the Original Test Data for the Taylor Standard Series, The David W. Taylor Model Basin, Report 806, March 1954

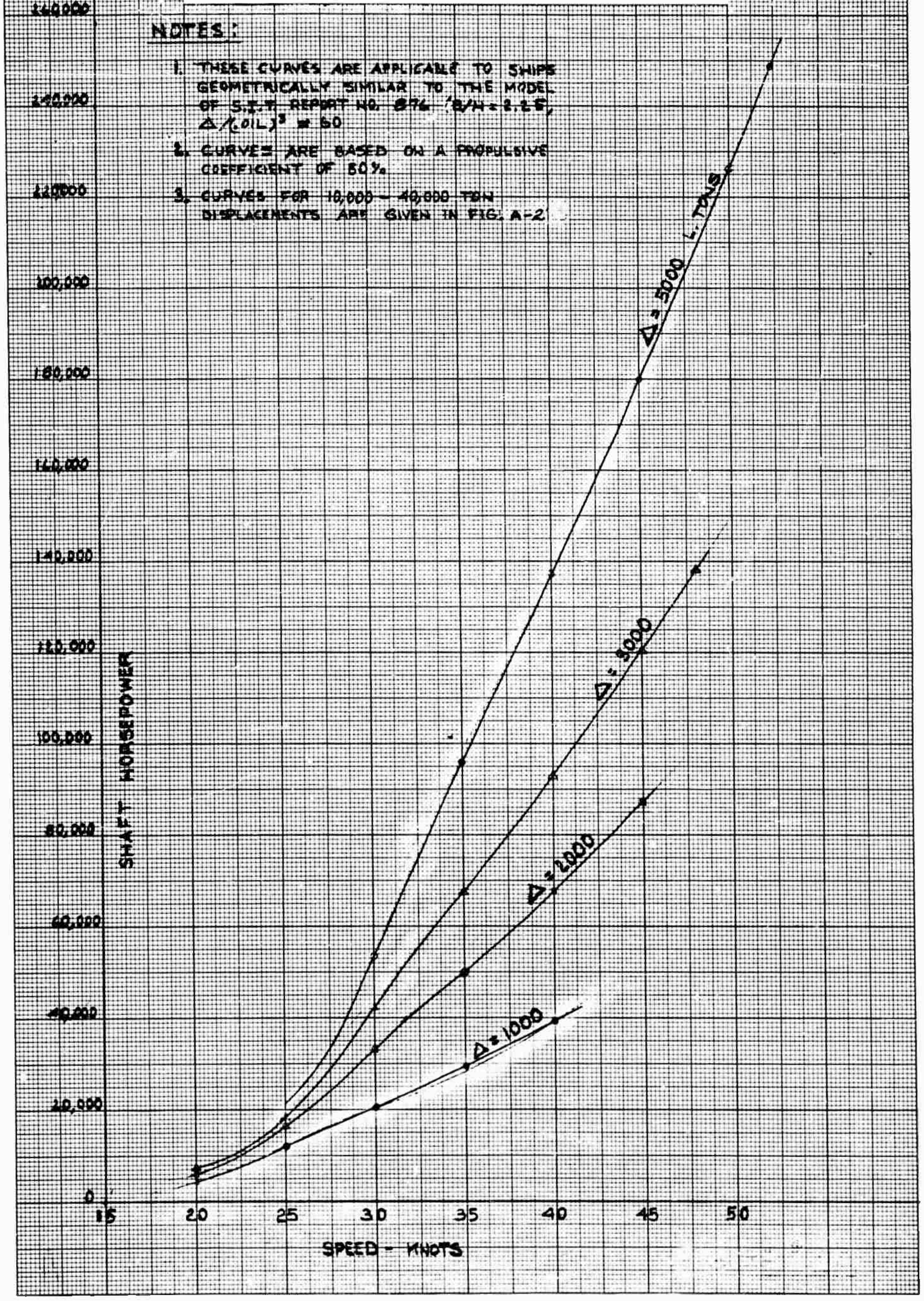
3. Principles of Naval Architecture, edited by Henry E. Rossell and Lawrence B. Chapman, published by SNAME

4. Uniform Procedure for the Calculation of Frictional Resistance and the Expansion of Model Test Data to Full Size, SNAME, Technical and Research Bulletin No. 1-2, August 1948

5. Sutherland, William H., Feasibility and Costs of High-Speed Ships for Strategic Deployment of Army Forces, Operation Research Office, The Johns Hopkins University, Staff Paper ORO-SP-131, April 1960

6. Johnson, Roger P. and Henry P. Rumble, Weight, Cost and Design Characteristics of Tankers and Dry Cargo Ships, The Rand Corporation, Memorandum RM-3318-PR, April 1963

FIG. A-1 ADVANCED DISPLACEMENT HULLS, SHAFT HORSEPOWER VERSUS SPEED
(Displacement, 1,000 to 5,000 Tons)



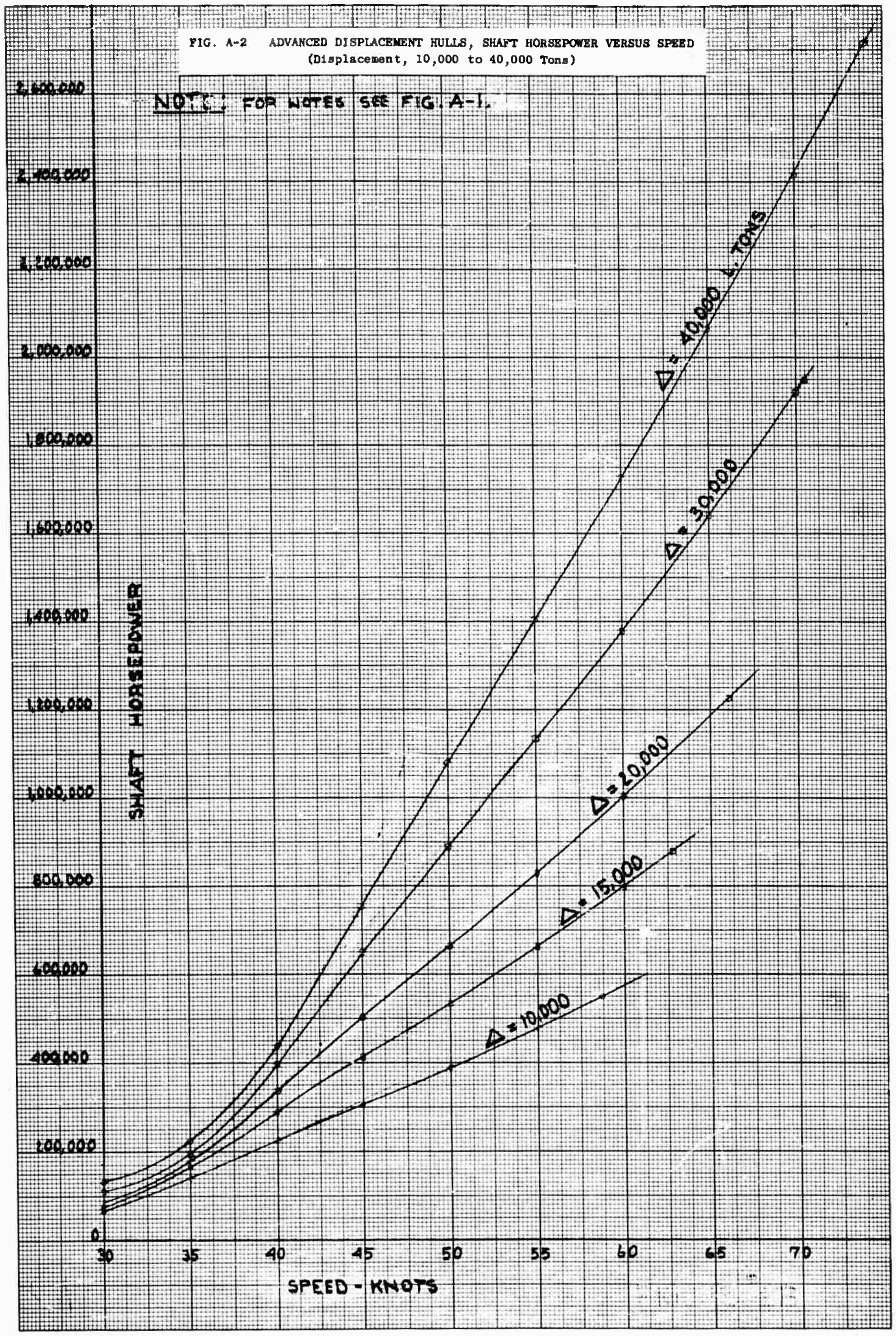


FIG. A-2 ADVANCED DISPLACEMENT HULLS, SHAFT HORSEPOWER VERSUS SPEED
(Displacement, 10,000 to 40,000 Tons)
NOTE: FOR NOTES SEE FIG. A-1.
SHAFT HORSEPOWER
SPEED - KNOTS
Δ = 40,000 L. TONS
Δ = 30,000
Δ = 20,000
Δ = 15,000
Δ = 10,000
2,600,000
2,400,000
2,200,000
2,000,000
1,800,000
1,600,000
1,400,000
1,200,000
1,000,000
800,000
600,000
400,000
200,000
0
30
35
40
45
50
55
60
65
70

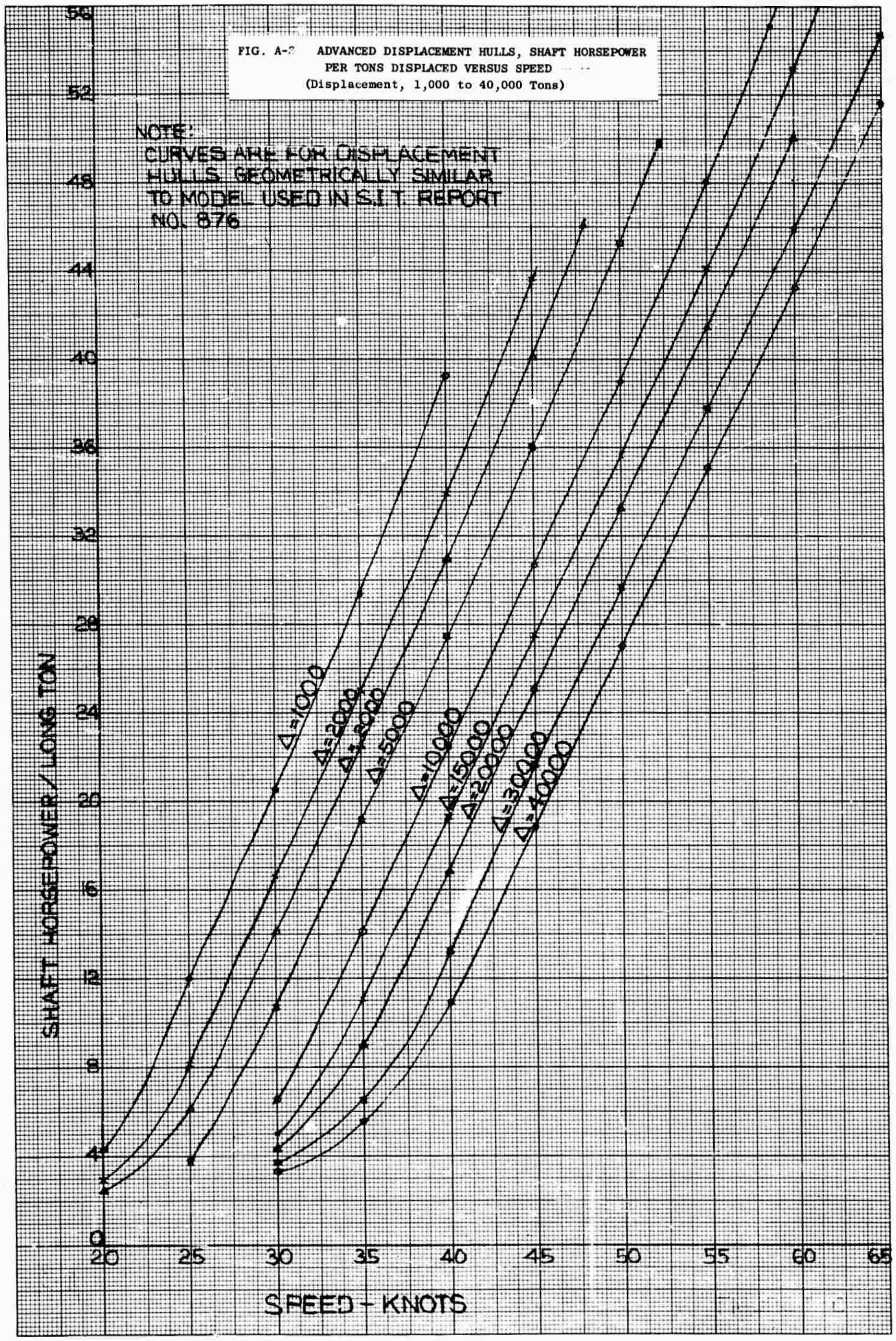


FIG. A-? ADVANCED DISPLACEMENT HULLS, SHAFT HORSEPOWER PER TONS DISPLACED VERSUS SPEED
(Displacement, 1,000 to 40,000 Tons)

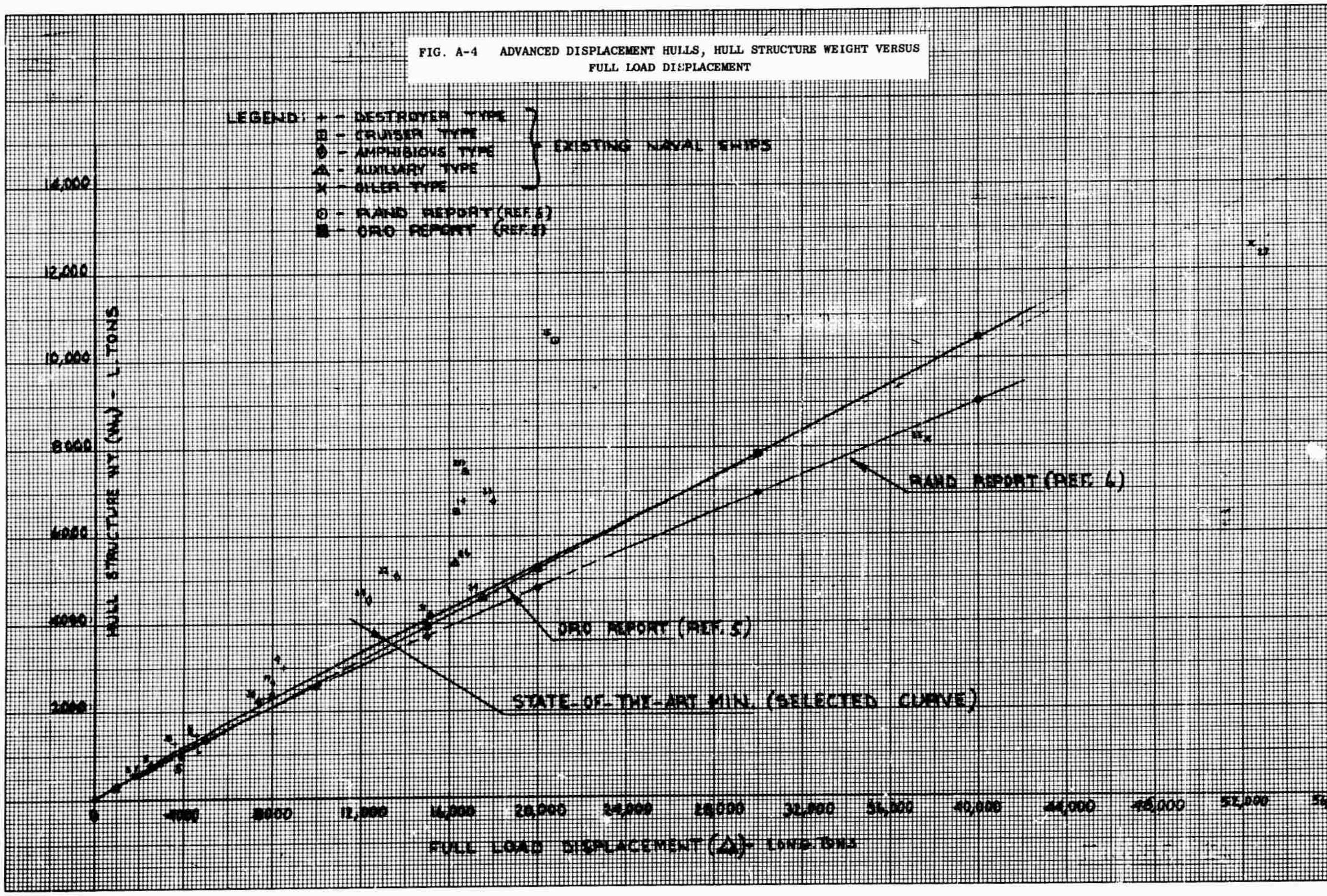


FIG. A-4 ADVANCED DISPLACEMENT HULLS, HULL STRUCTURE WEIGHT VERSUS FULL LOAD DISPLACEMENT

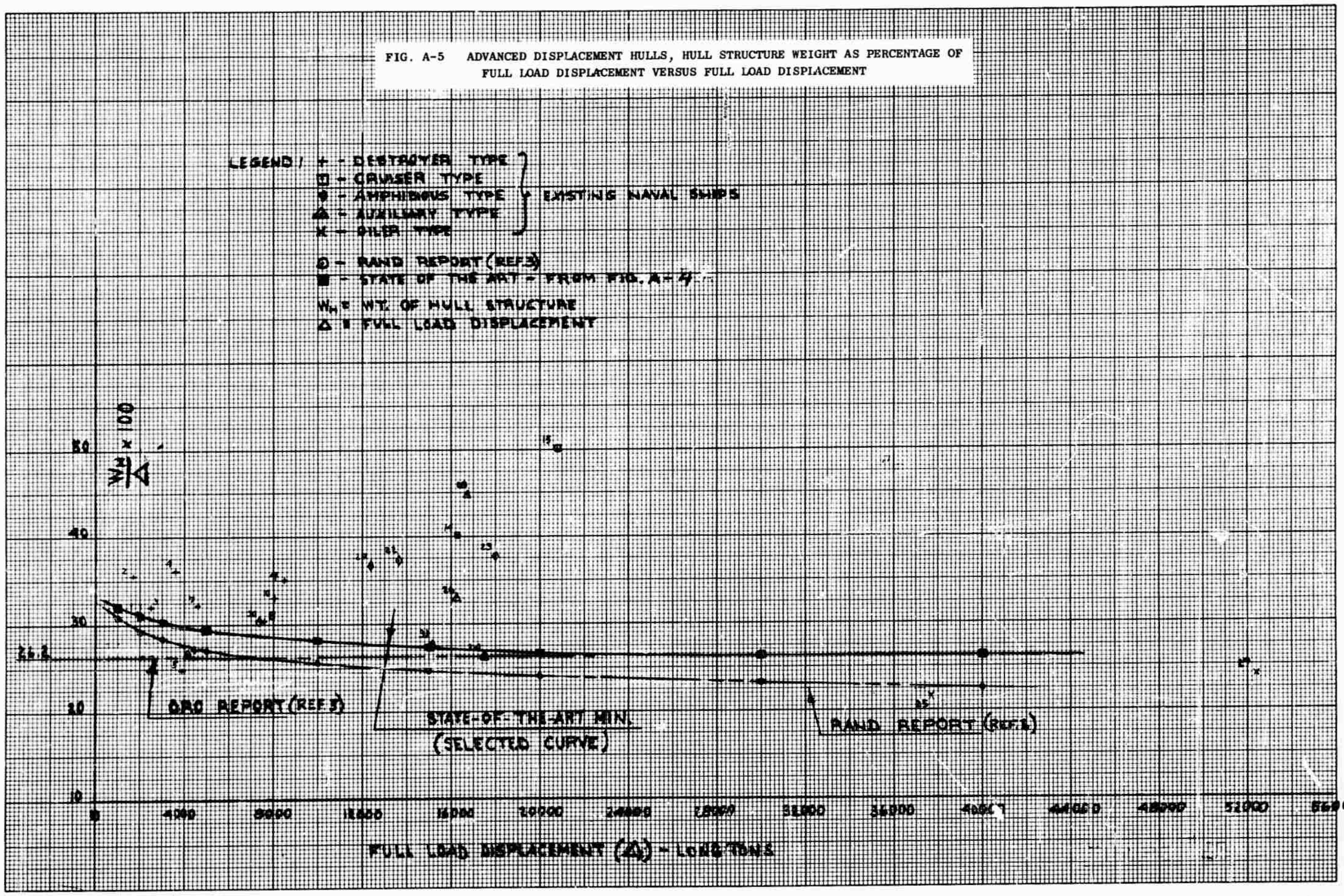


FIG. A-5 ADVANCED DISPLACEMENT HULLS, HULL STRUCTURE WEIGHT AS PERCENTAGE OF FULL LOAD DISPLACEMENT VERSUS FULL LOAD DISPLACEMENT

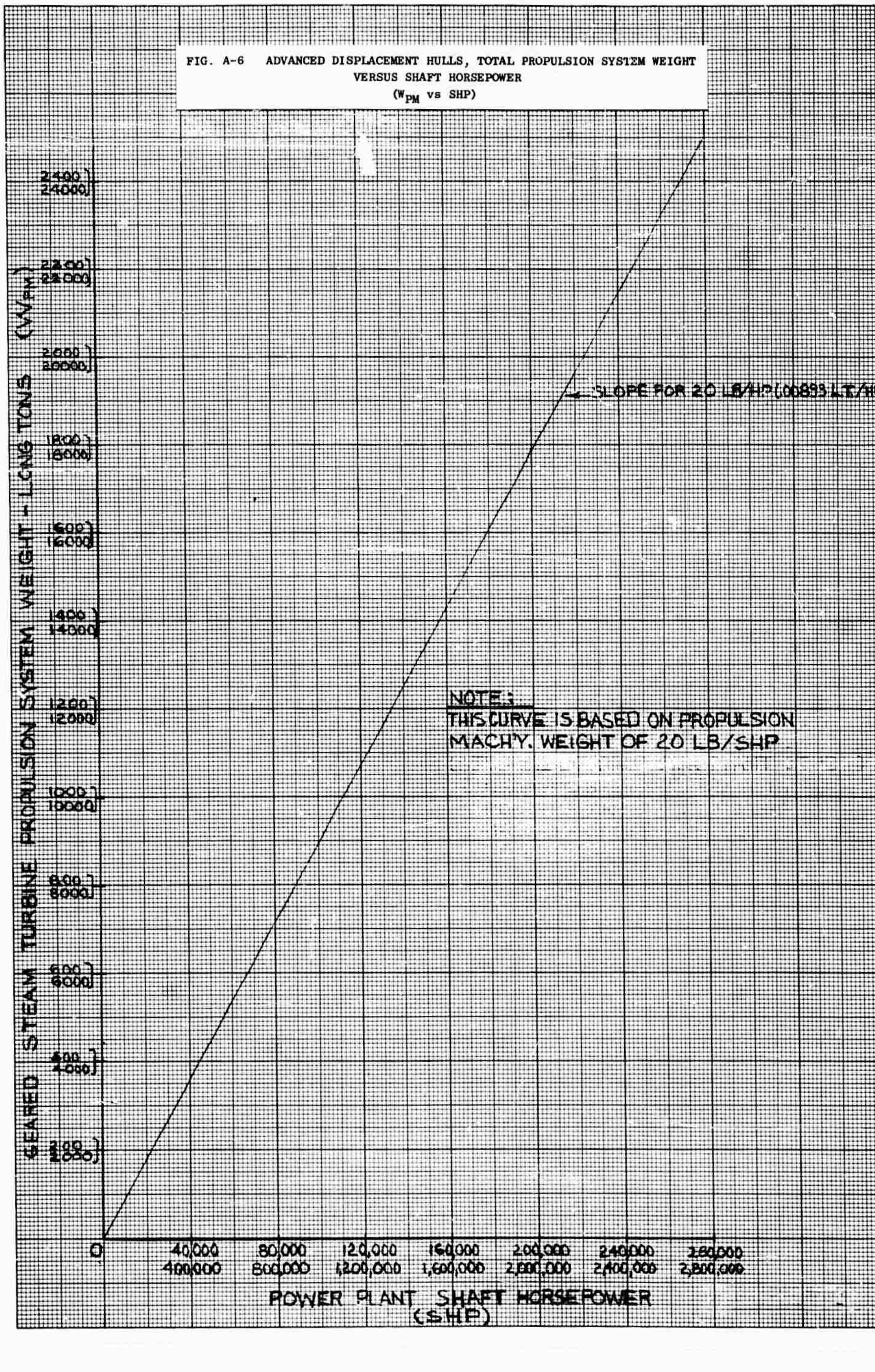
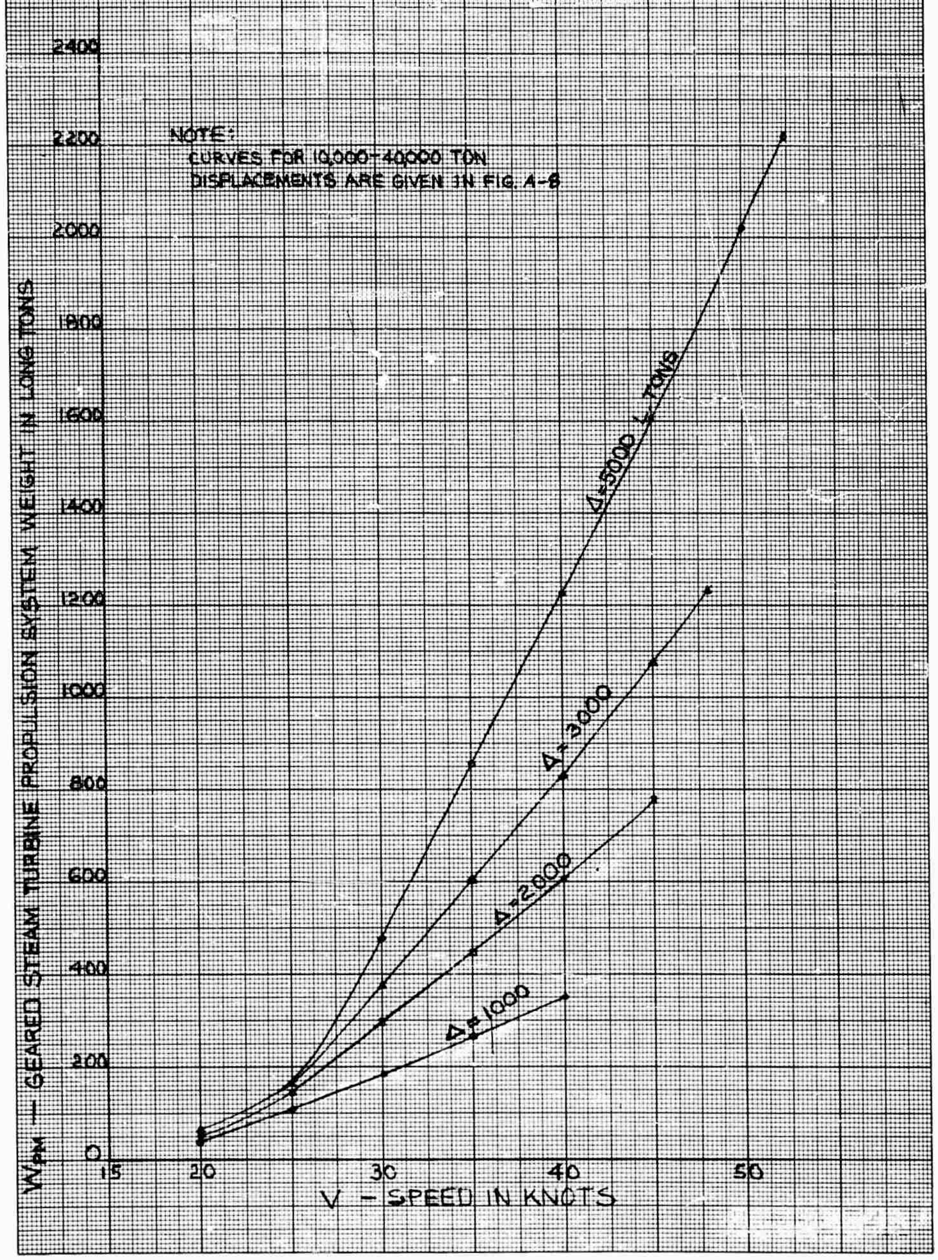


FIG. A-6 ADVANCED DISPLACEMENT HULLS, TOTAL PROPULSION SYSTEM WEIGHT VERSUS SHAFT HORSEPOWER (W_{PM} vs SHP)

FIG. A-7 ADVANCED DISPLACEMENT HULLS, PROPULSION SYSTEM WEIGHT VERSUS SPEED (W_{PM} vs V)
(Displacement, 1,000 to 5,000 Tons)



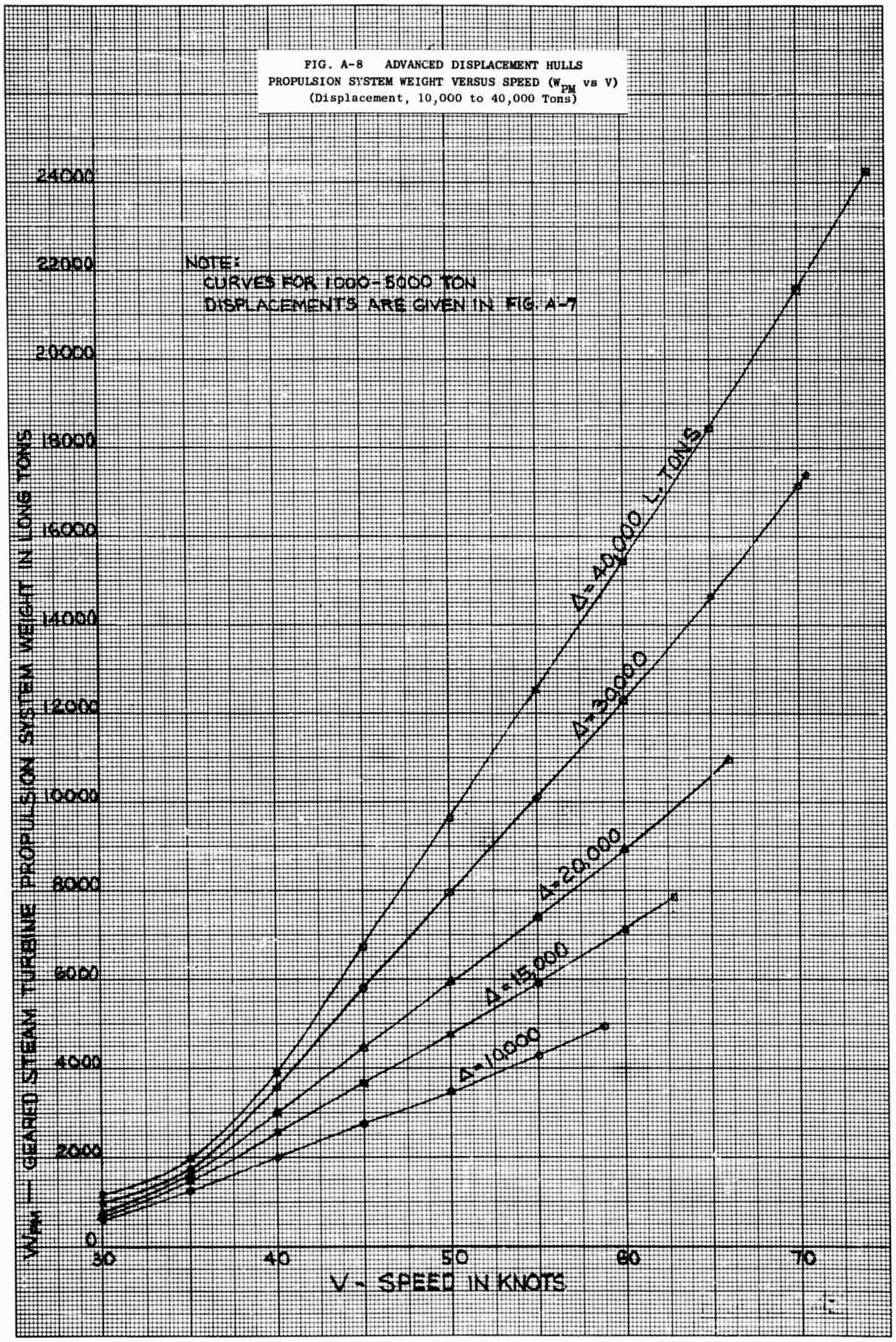


FIG. A-8 ADVANCED DISPLACEMENT HULLS
PROPULSION SYSTEM WEIGHT VERSUS SPEED (W_{PM} vs V)
(Displacement, 10,000 to 40,000 Tons)
NOTE:
CURVES FOR 1000-5000 TON
DISPLACEMENTS ARE GIVEN IN FIG. A-7
W_{PM} — GEARED STEAM TURBINE PROPULSION SYSTEM WEIGHT IN LONG TONS
V - SPEED IN KNOTS
Δ=40,000 L. TONS
Δ=30,000
Δ=20,000
Δ=15,000
Δ=10,000
0
2000
4000
6000
8000
10000
12000
14000
16000
18000
20000
22000
24000
30
40
50
60
70

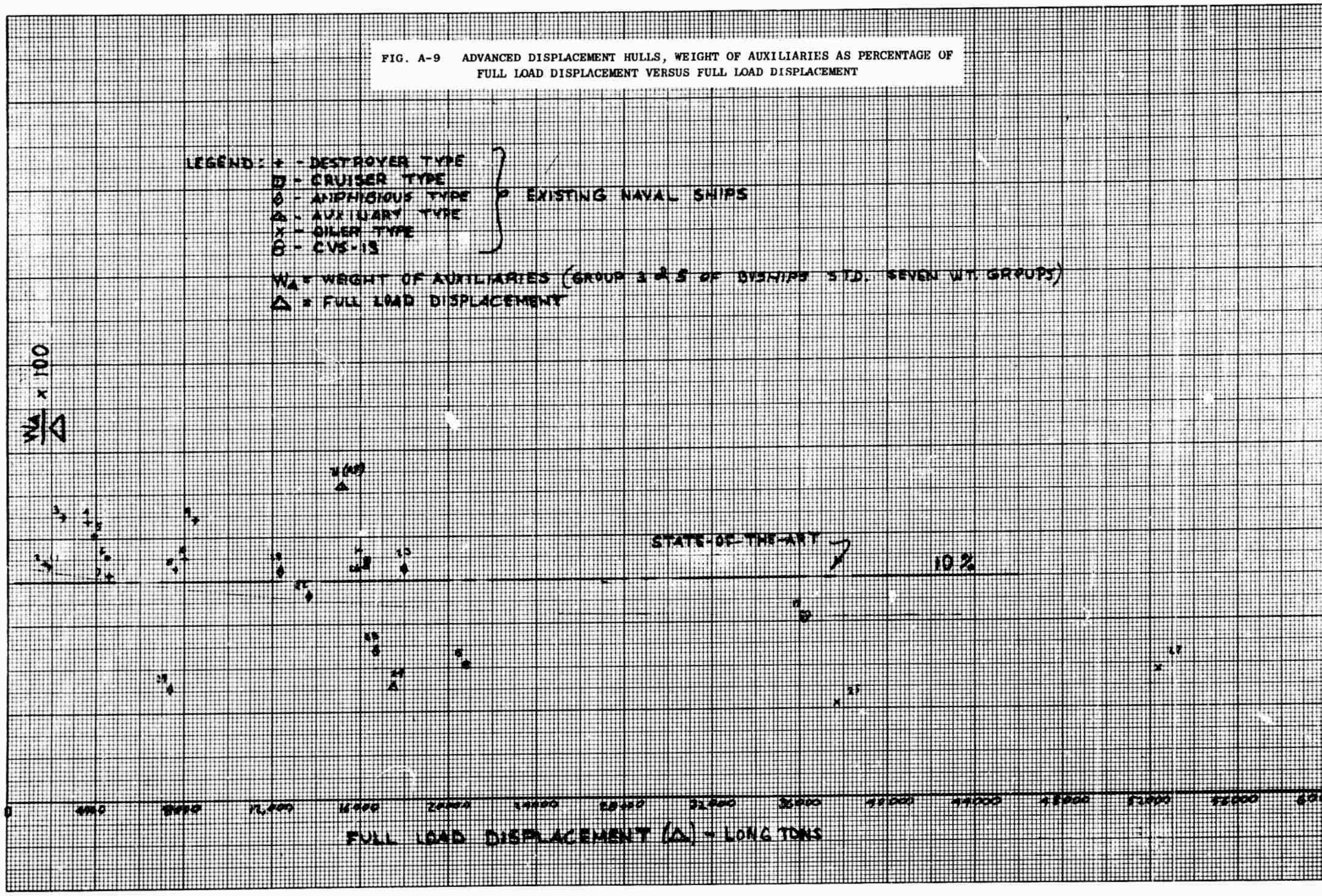


FIG. A-9 ADVANCED DISPLACEMENT HULLS, WEIGHT OF AUXILIARIES AS PERCENTAGE OF FULL LOAD DISPLACEMENT VERSUS FULL LOAD DISPLACEMENT

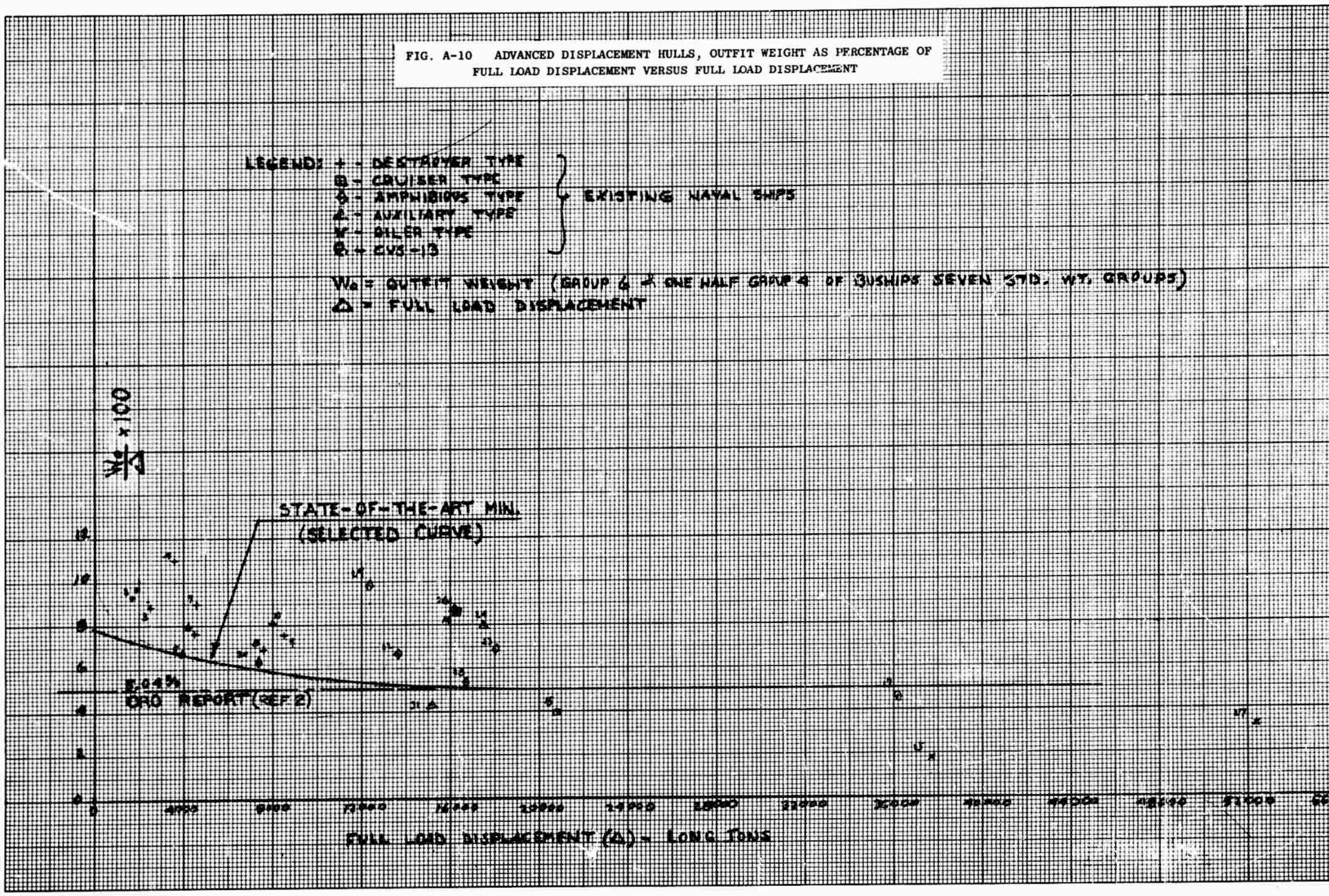
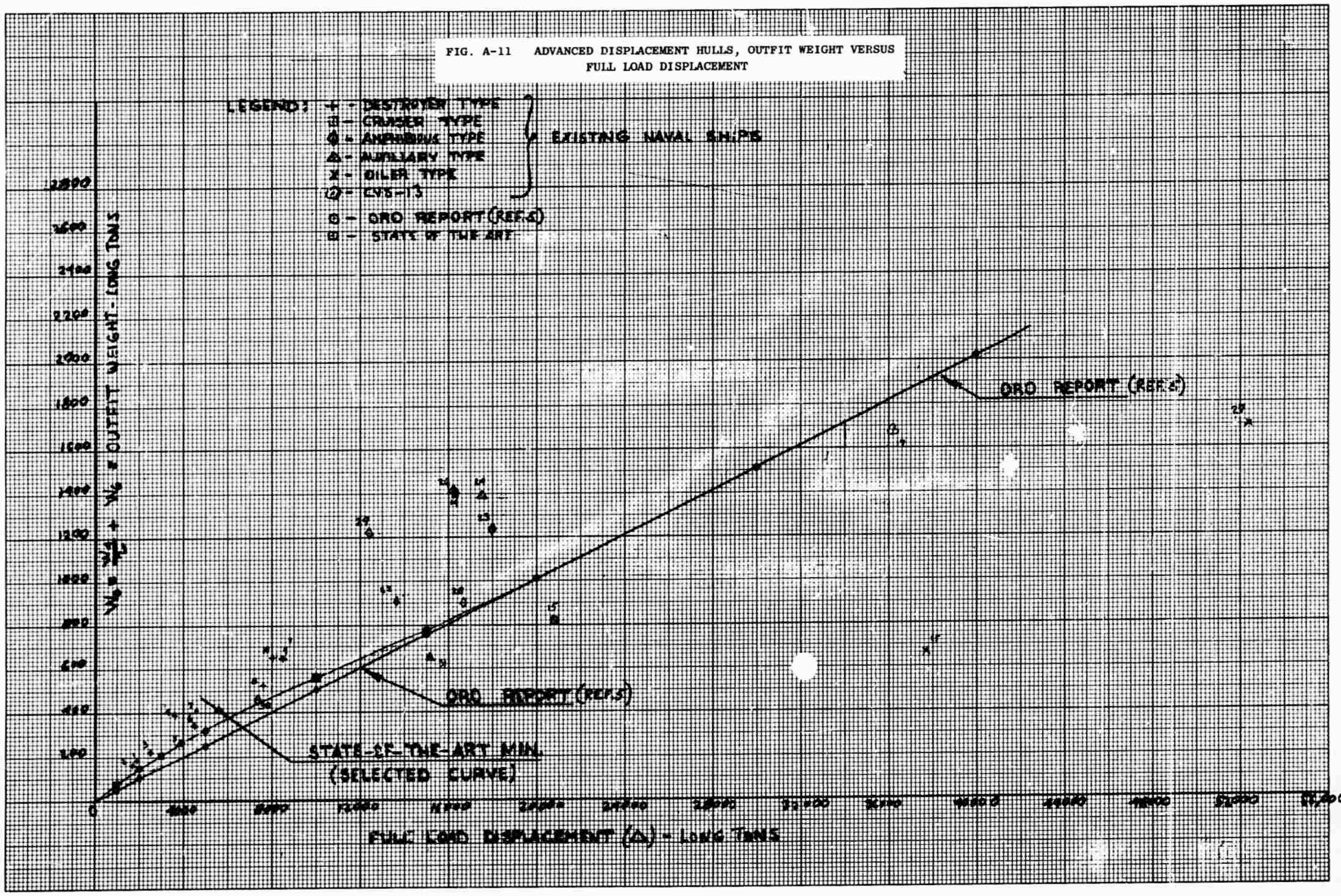


FIG. A-10 ADVANCED DISPLACEMENT HULLS, OUTFIT WEIGHT AS PERCENTAGE OF FULL LOAD DISPLACEMENT VERSUS FULL LOAD DISPLACEMENT

FIG. A-11 ADVANCED DISPLACEMENT HULLS, OUTFIT WEIGHT VERSUS FULL LOAD DISPLACEMENT



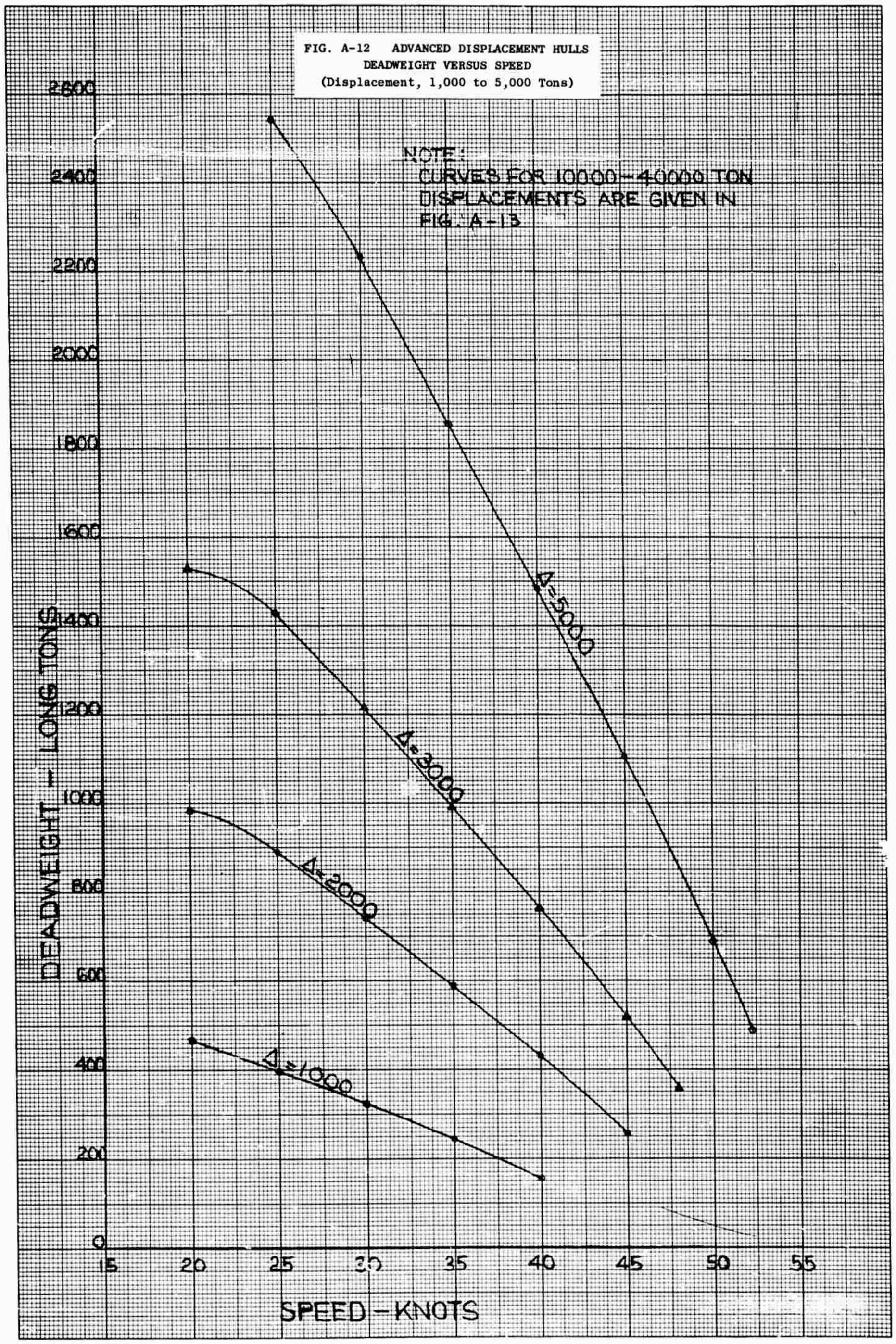
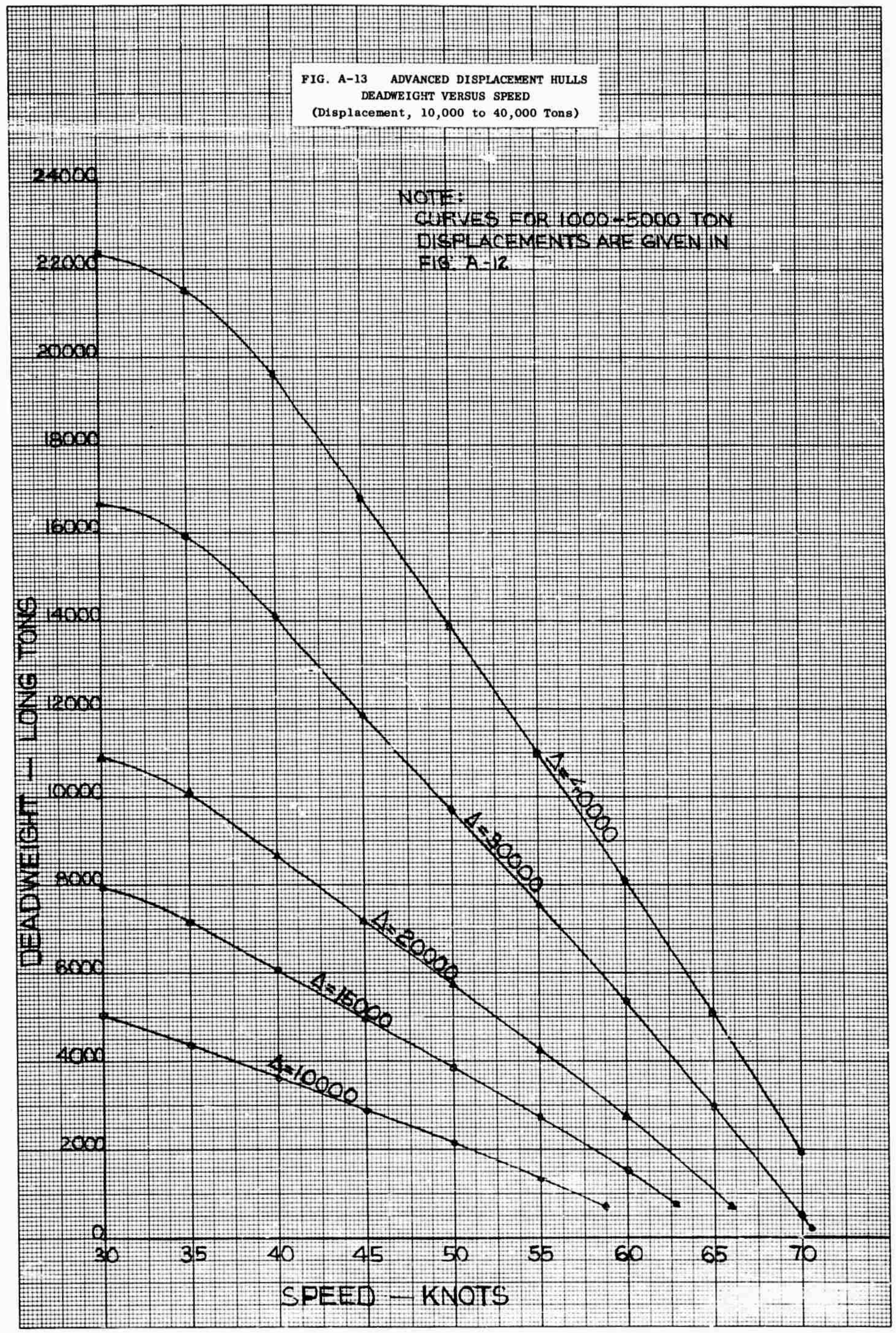


FIG. A-12 ADVANCED DISPLACEMENT HULLS
DEADWEIGHT VERSUS SPEED
(Displacement, 1,000 to 5,000 Tons)
NOTE:
CURVES FOR 10000-40000 TON DISPLACEMENTS ARE GIVEN IN FIG. A-13
DEADWEIGHT - LONG TONS
SPEED - KNOTS
Δ=5000
Δ=3000
Δ=2000
Δ=1000
0
200
400
600
800
1000
1200
1400
1600
1800
2000
2200
2400
2600
15
20
25
30
35
40
45
50
55

FIG. A-13 ADVANCED DISPLACEMENT HULLS
DEADWEIGHT VERSUS SPEED
(Displacement, 10,000 to 40,000 Tons)



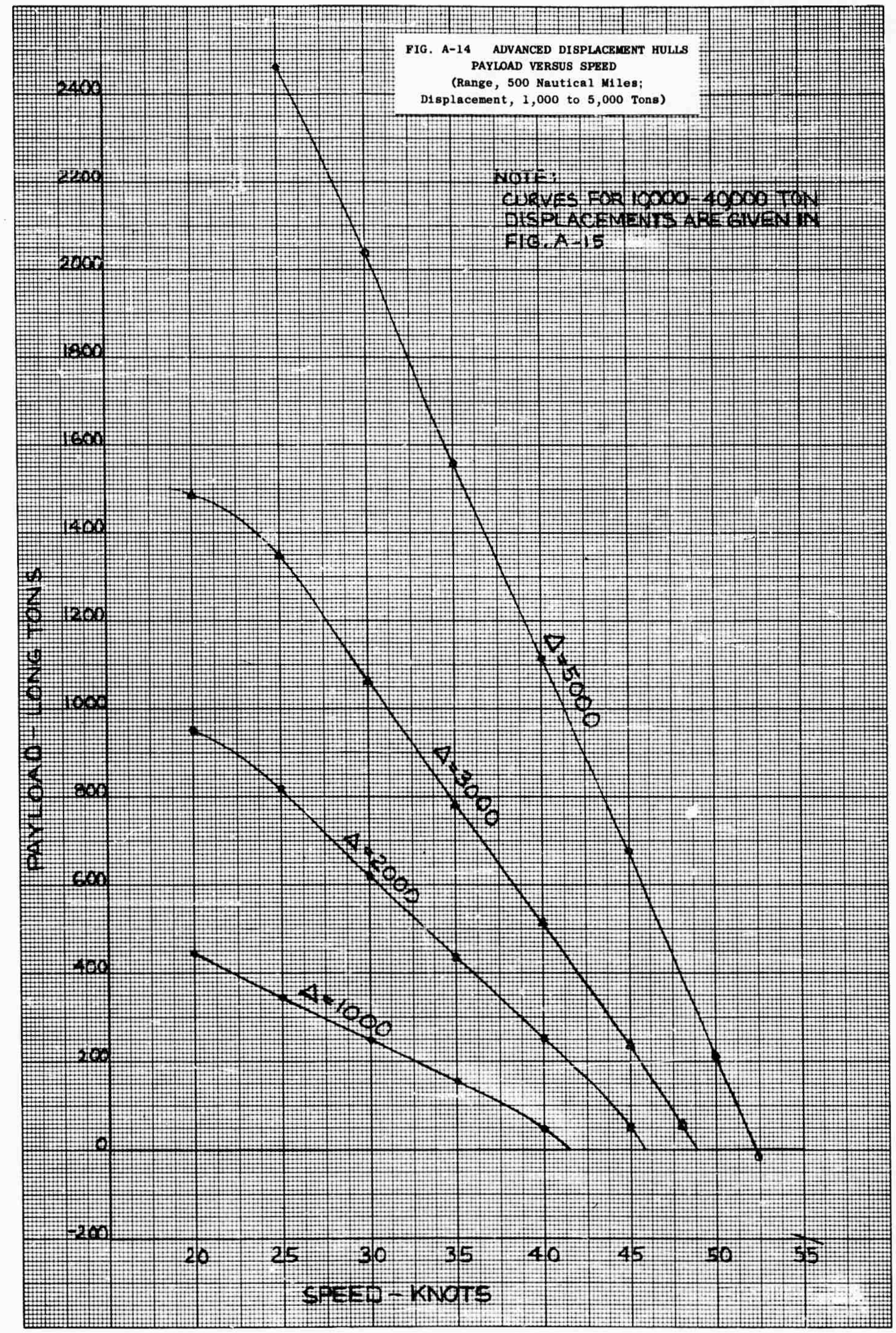


FIG. A-14 ADVANCED DISPLACEMENT HULLS
PAYLOAD VERSUS SPEED
(Range, 500 Nautical Miles;
Displacement, 1,000 to 5,000 Tons)
NOTE:
CURVES FOR 10000-40000 TON
DISPLACEMENTS ARE GIVEN IN
FIG. A-15
PAYLOAD - LONG TONS
SPEED - KNOTS
Δ=5000
Δ=3000
Δ=2000
Δ=1000
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
0
-200
20
25
30
35
40
45
50
55

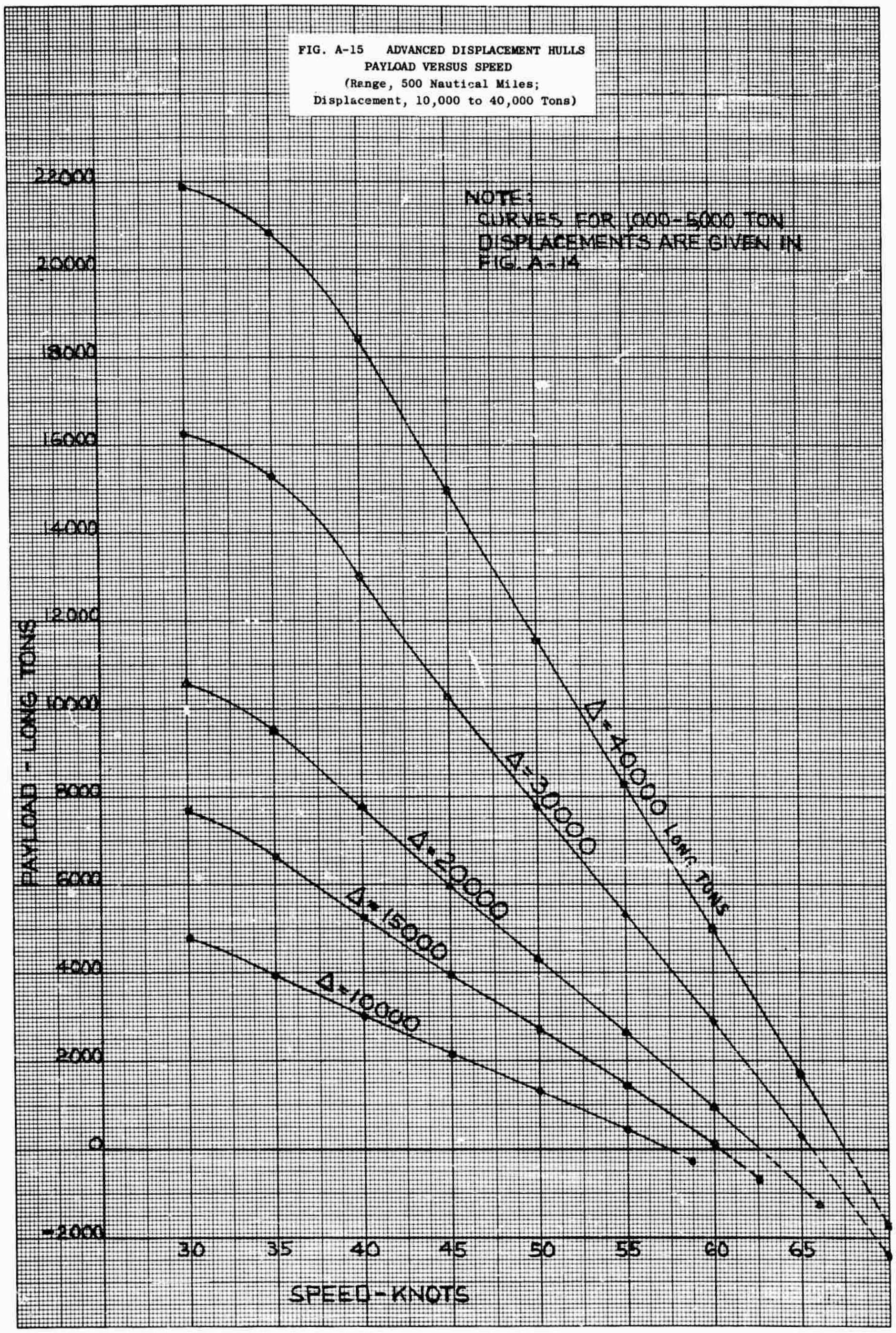


FIG. A-15 ADVANCED DISPLACEMENT HULLS
PAYLOAD VERSUS SPEED
(Range, 500 Nautical Miles;
Displacement, 10,000 to 40,000 Tons)
NOTE:
CURVES FOR 1000-5000 TON
DISPLACEMENTS ARE GIVEN IN
FIG. A-14
PAYLOAD - LONG TONS
SPEED - KNOTS
22000
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
-2000
30
35
40
45
50
55
60
65
Δ=40000 LONG TONS
Δ=30000
Δ=20000
Δ=15000
Δ=10000

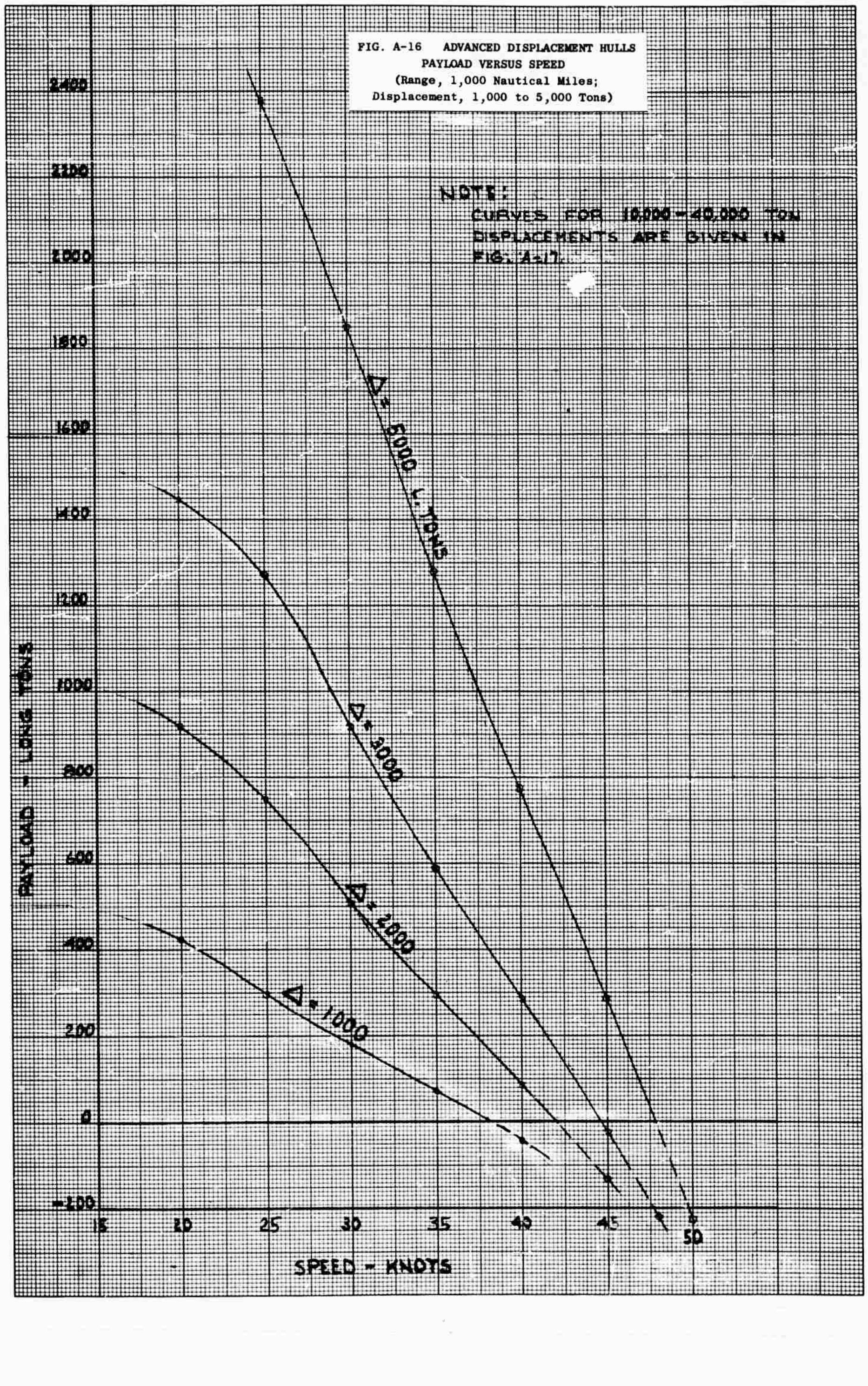


FIG. A-16 ADVANCED DISPLACEMENT HULLS
PAYLOAD VERSUS SPEED
(Range, 1,000 Nautical Miles;
Displacement, 1,000 to 5,000 Tons)

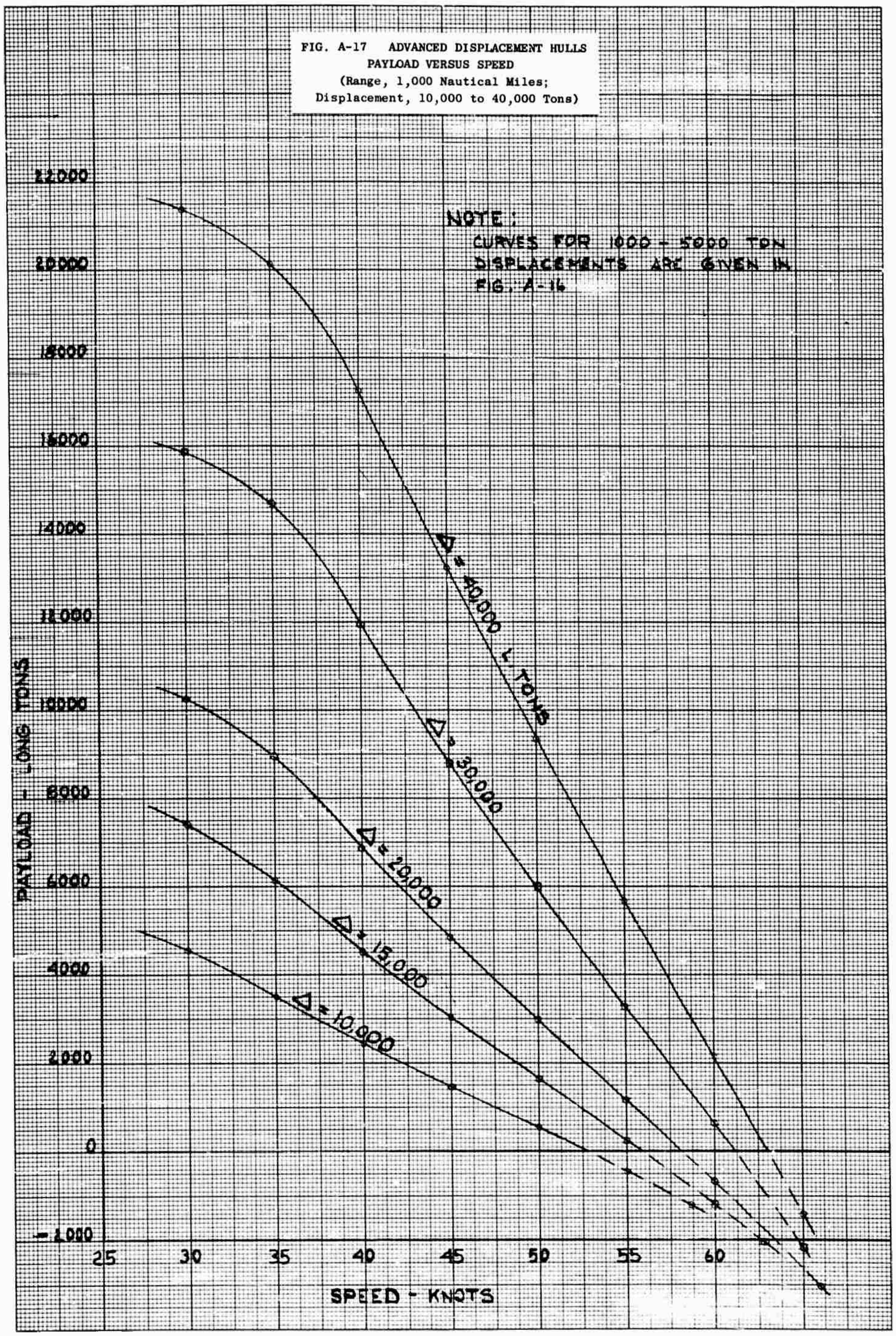


FIG. A-17 ADVANCED DISPLACEMENT HULLS
PAYLOAD VERSUS SPEED
(Range, 1,000 Nautical Miles;
Displacement, 10,000 to 40,000 Tons)
NOTE:
CURVES FOR 1000 - 5000 TON
DISPLACEMENTS ARE GIVEN IN
FIG. A-16
Δ = 40,000 L. TONS
Δ = 30,000
Δ = 20,000
Δ = 15,000
Δ = 10,000
PAYLOAD - LONG TONS
22000
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
-1000
25
30
35
40
45
50
55
60
SPEED - KNOTS

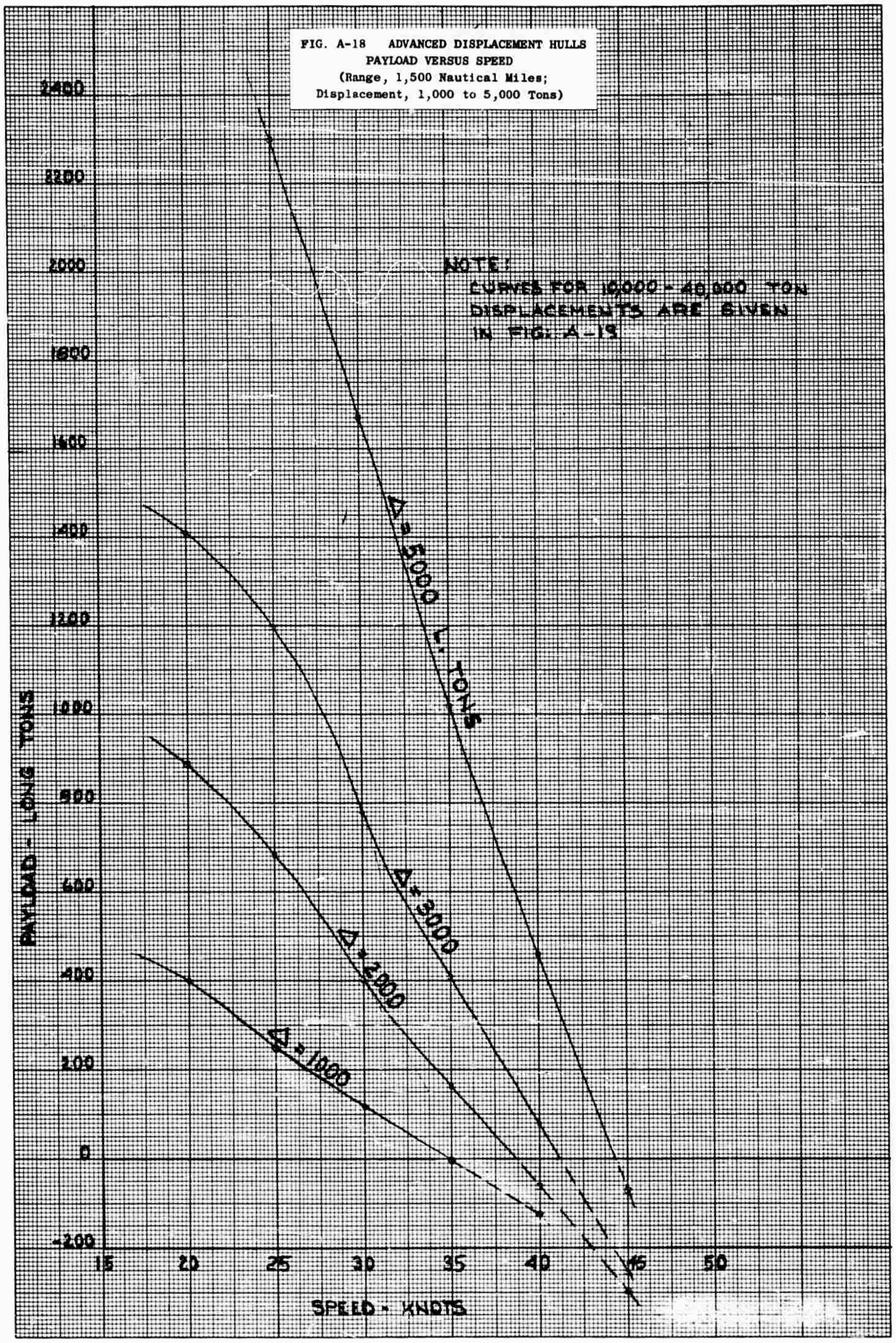


FIG. A-18 ADVANCED DISPLACEMENT HULLS
PAYLOAD VERSUS SPEED
(Range, 1,500 Nautical Miles;
Displacement, 1,000 to 5,000 Tons)

FIG. A-19 ADVANCED DISPLACEMENT HULLS
PAYLOAD VERSUS SPEED
(Range, 1,500 Nautical Miles;
Displacement, 10,000 to 40,000 Tons)

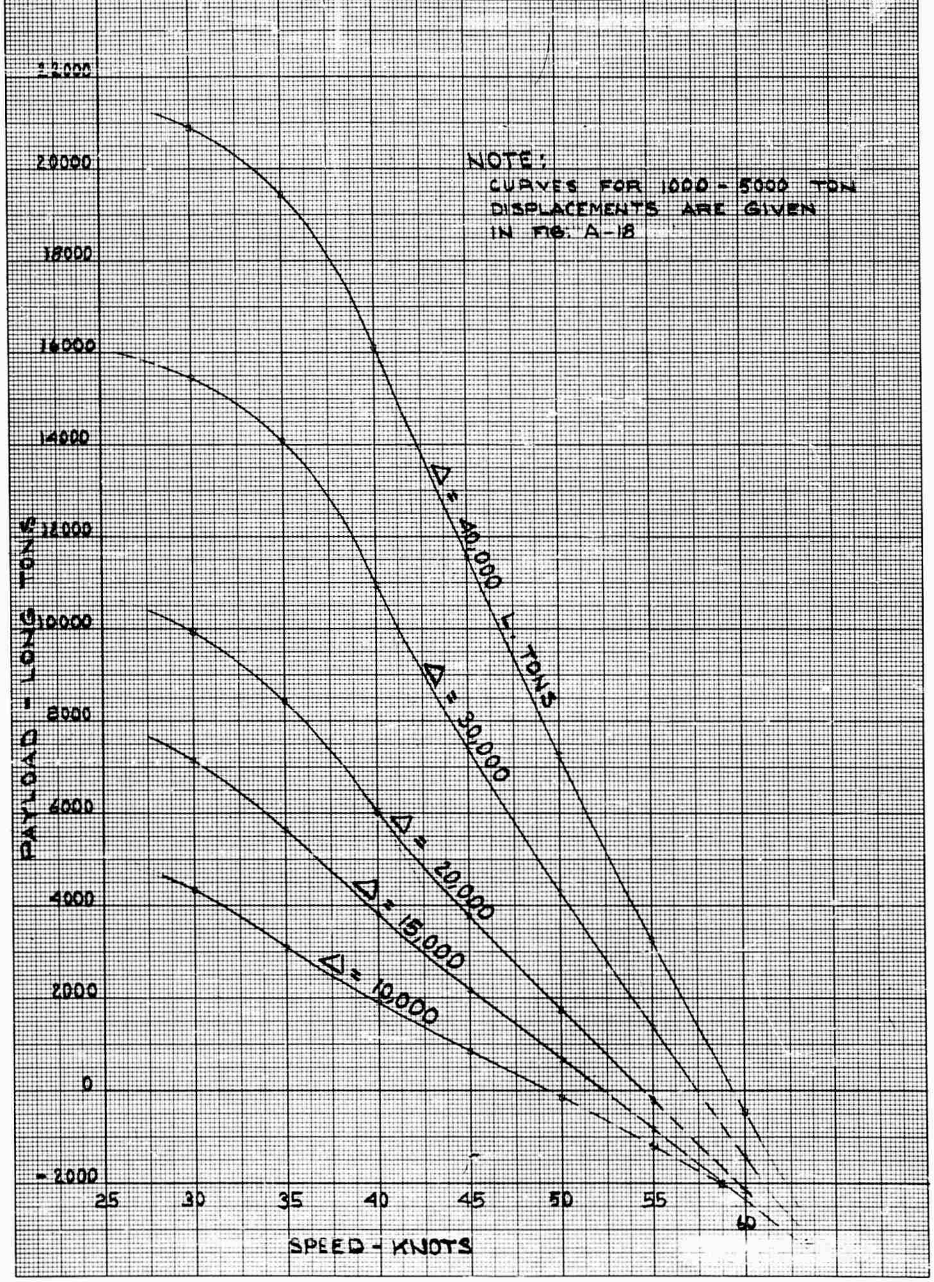
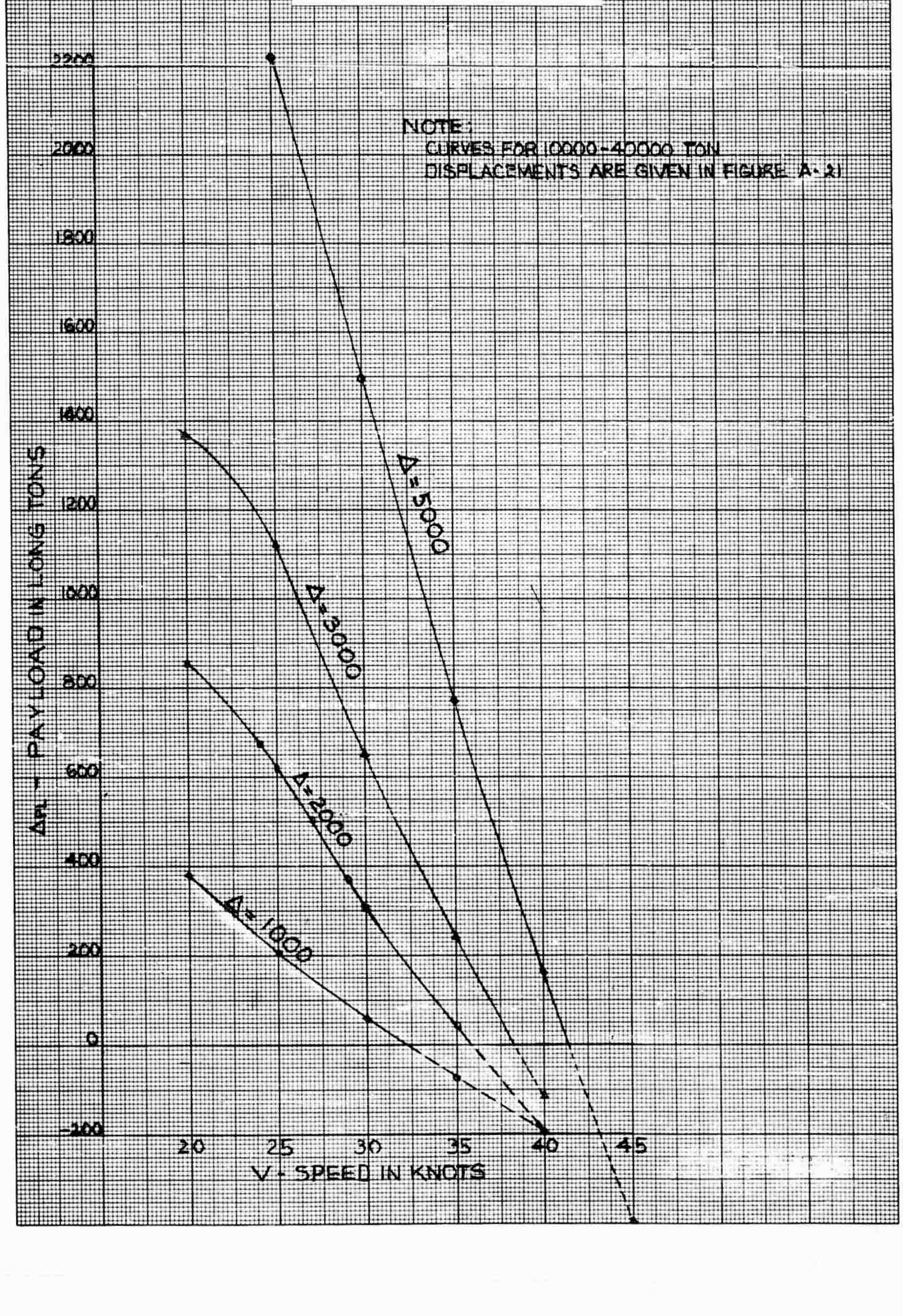


FIG. A-20 ADVANCED DISPLACEMENT HULLS
PAYLOAD VERSUS SPEED
(Range, 2,000 Nautical Miles;
Displacement, 1,000 to 5,000 Tons)



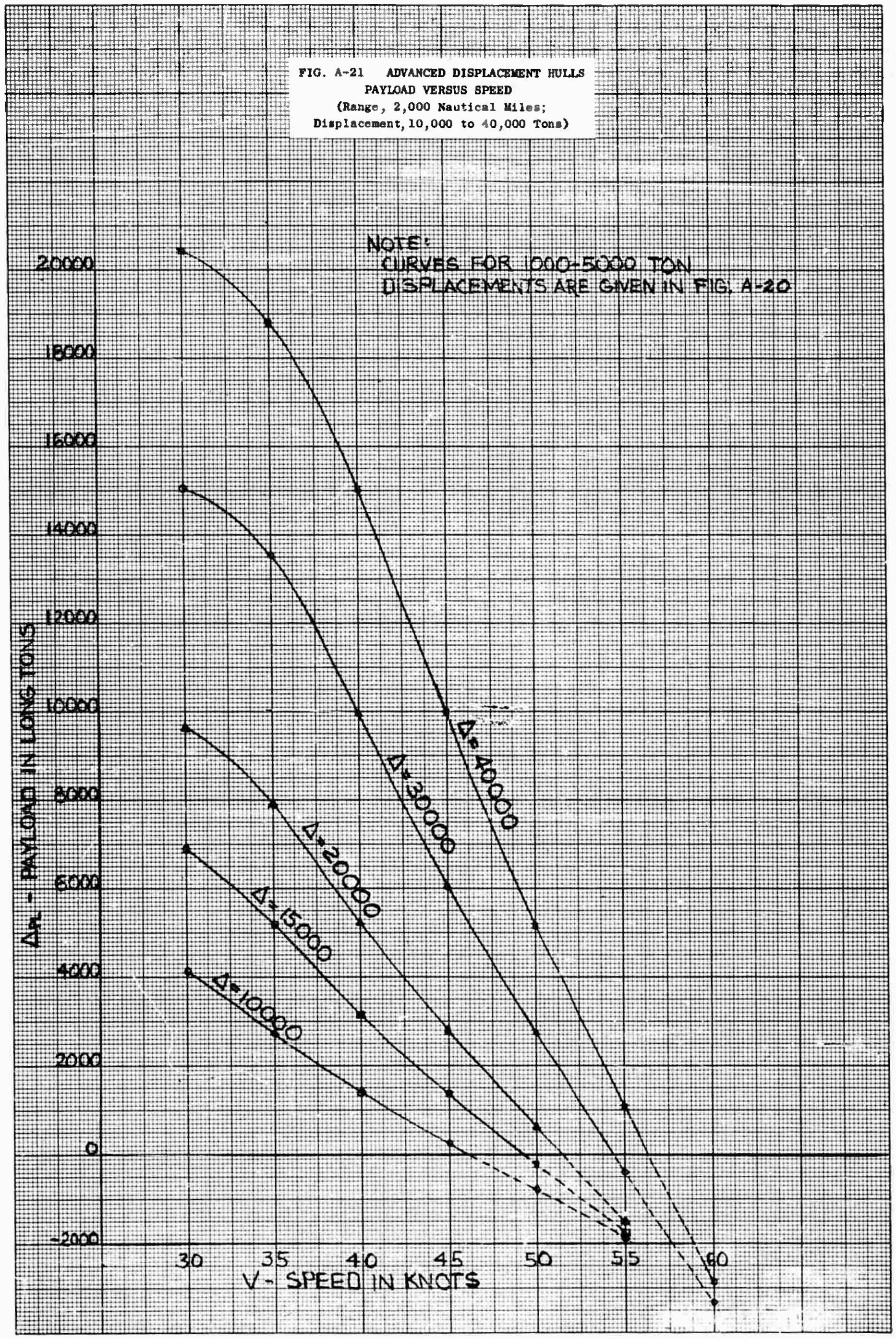


FIG. A-21 ADVANCED DISPLACEMENT HULLS
PAYLOAD VERSUS SPEED
(Range, 2,000 Nautical Miles;
Displacement, 10,000 to 40,000 Tons)
NOTE:
CURVES FOR 1000-5000 TON
DISPLACEMENTS ARE GIVEN IN FIG. A-20
Δ = 40000
Δ = 30000
Δ = 20000
Δ = 15000
Δ = 10000
ΔPL - PAYLOAD IN LONG TONS
V - SPEED IN KNOTS
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
0
-2000
30
35
40
45
50
55
60

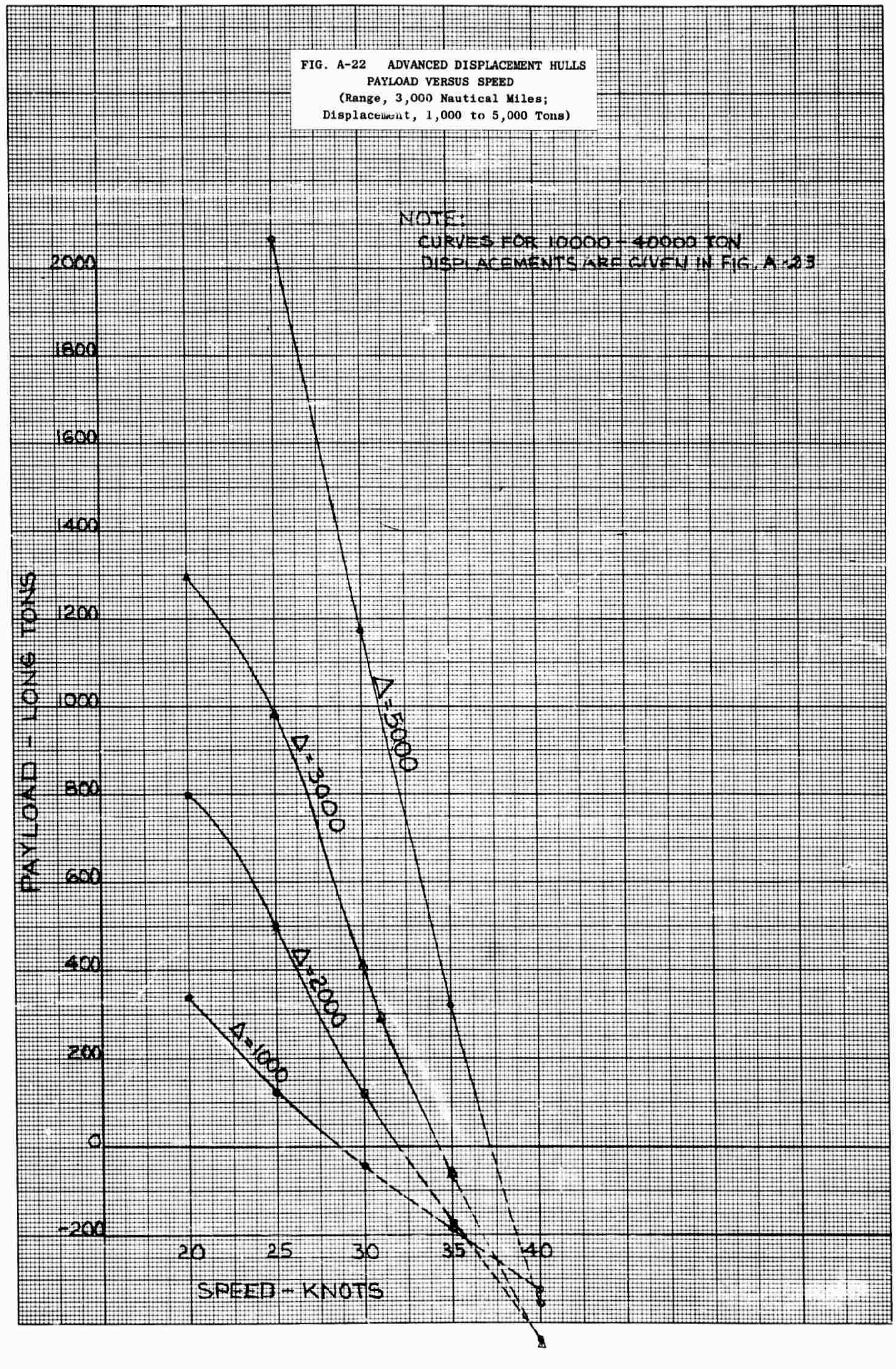


FIG. A-22 ADVANCED DISPLACEMENT HULLS
PAYLOAD VERSUS SPEED
(Range, 3,000 Nautical Miles;
Displacement, 1,000 to 5,000 Tons)
NOTE:
CURVES FOR 10000 - 40000 TON
DISPLACEMENTS ARE GIVEN IN FIG. A-23
PAYLOAD - LONG TONS
2000
1800
1600
1400
1200
1000
800
600
400
200
0
-200
20
25
30
35
40
SPEED - KNOTS
Δ=5000
Δ=3000
Δ=2000
Δ=1000

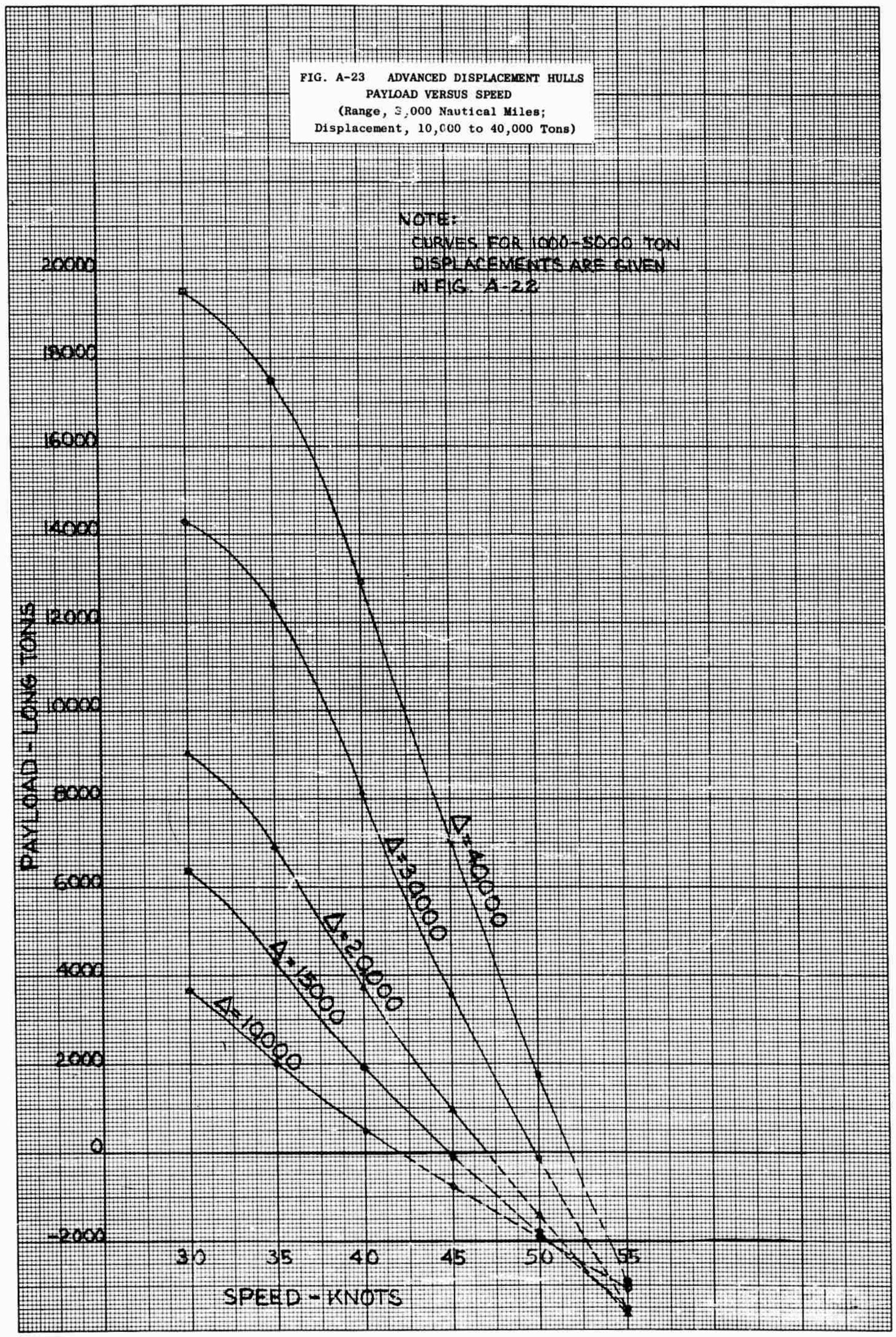


FIG. A-23 ADVANCED DISPLACEMENT HULLS
PAYLOAD VERSUS SPEED
(Range, 3,000 Nautical Miles;
Displacement, 10,000 to 40,000 Tons)

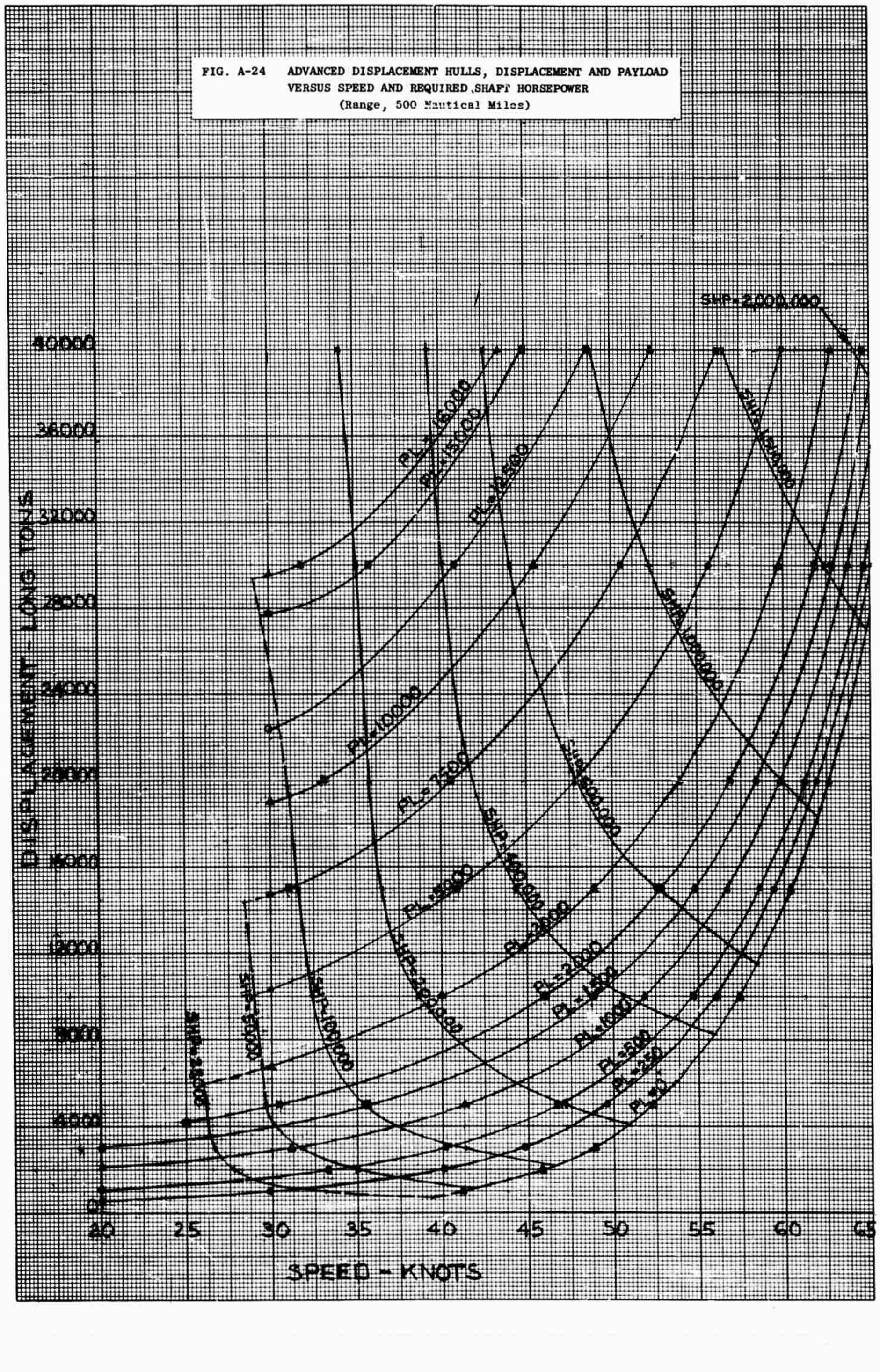


FIG. A-24 ADVANCED DISPLACEMENT HULLS, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER
(Range, 500 Nautical Miles)

FIG. A-25 ADVANCED DISPLACEMENT HULLS, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER
(Range, 1,000 Nautical Miles)

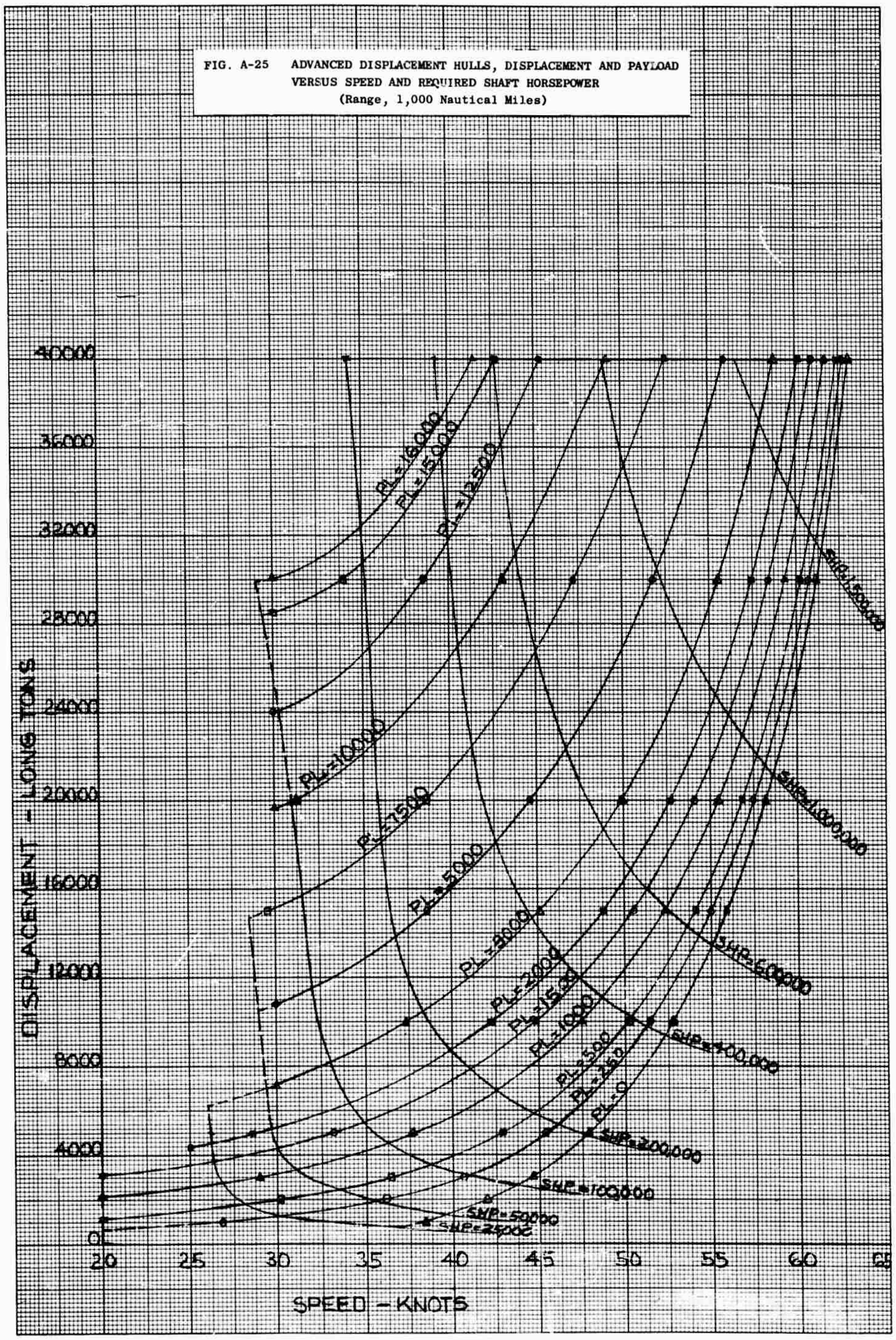
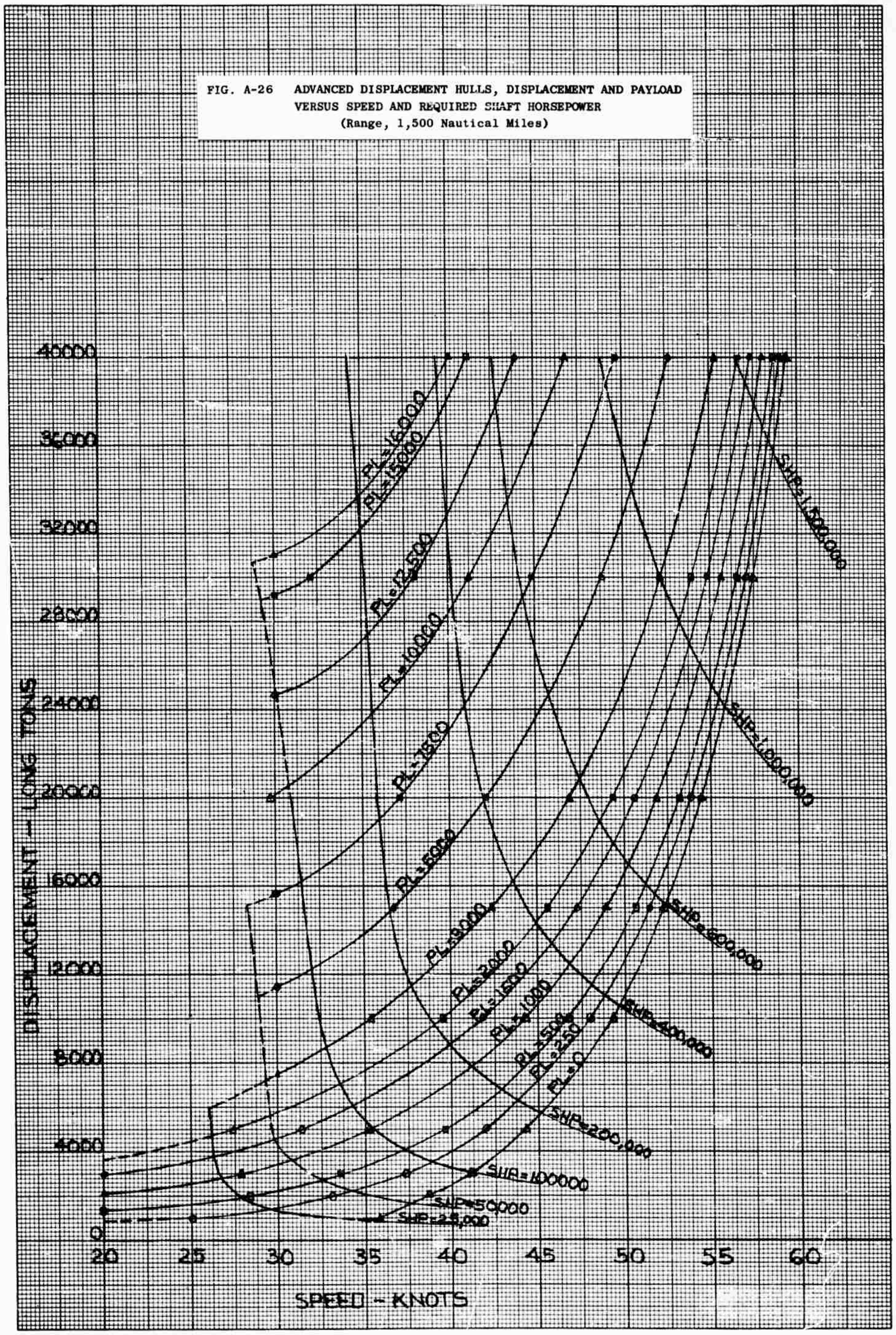


FIG. A-26 ADVANCED DISPLACEMENT HULLS, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER
(Range, 1,500 Nautical Miles)



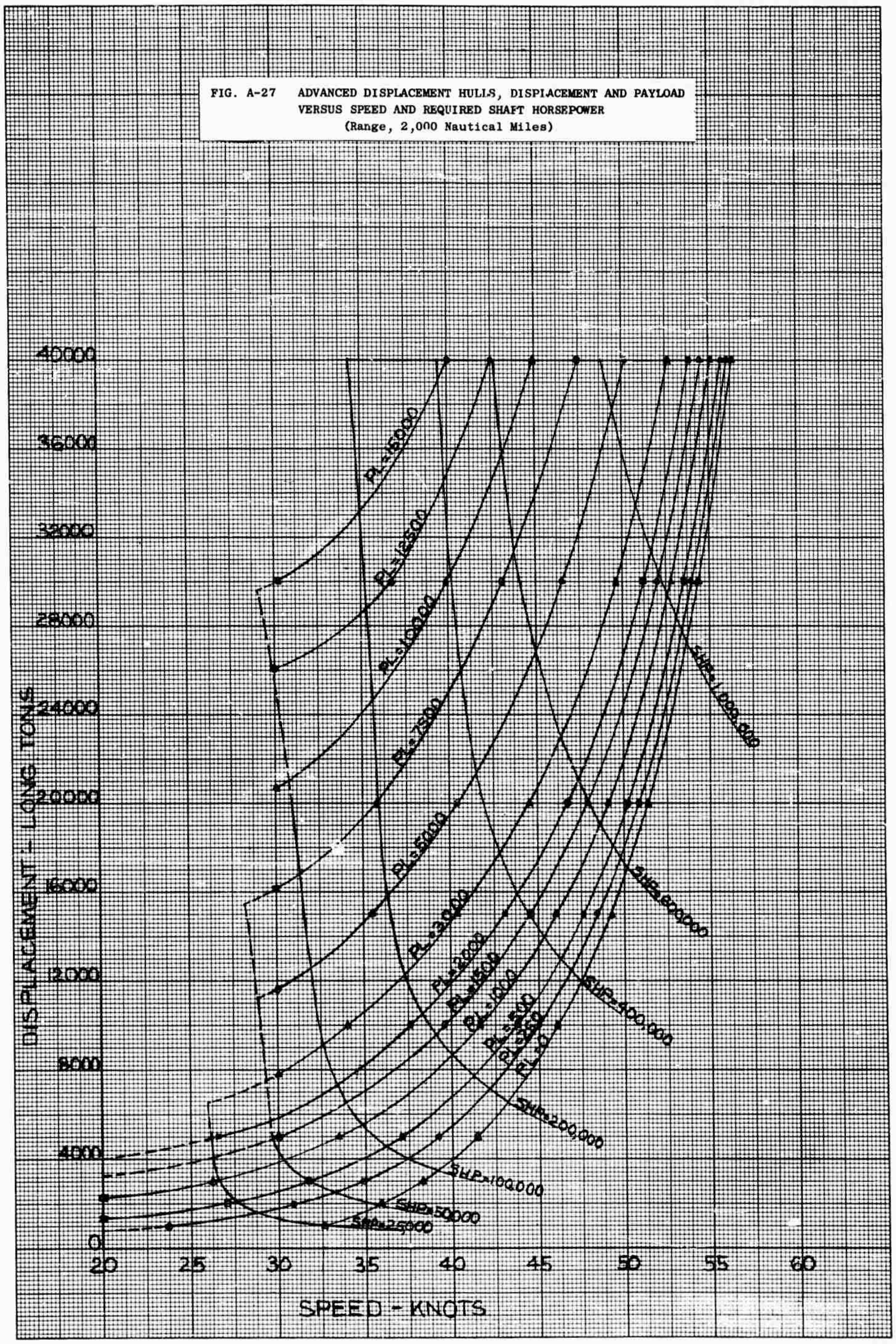


FIG. A-27 ADVANCED DISPLACEMENT HULLS, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER
(Range, 2,000 Nautical Miles)

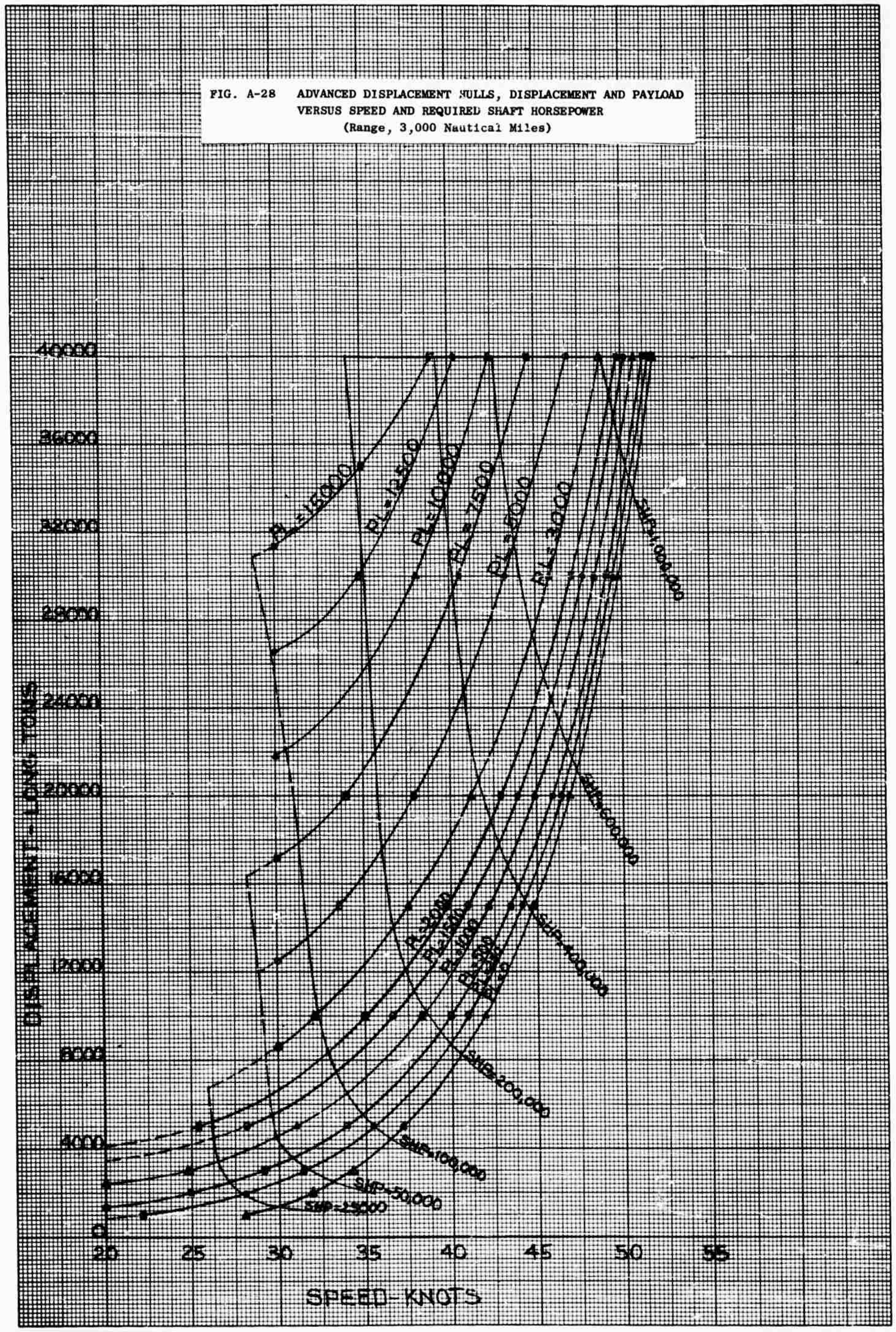


FIG. A-28 ADVANCED DISPLACEMENT HULLS, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER
(Range, 3,000 Nautical Miles)

FIG. A-29 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 25 Knots; Various Ranges)

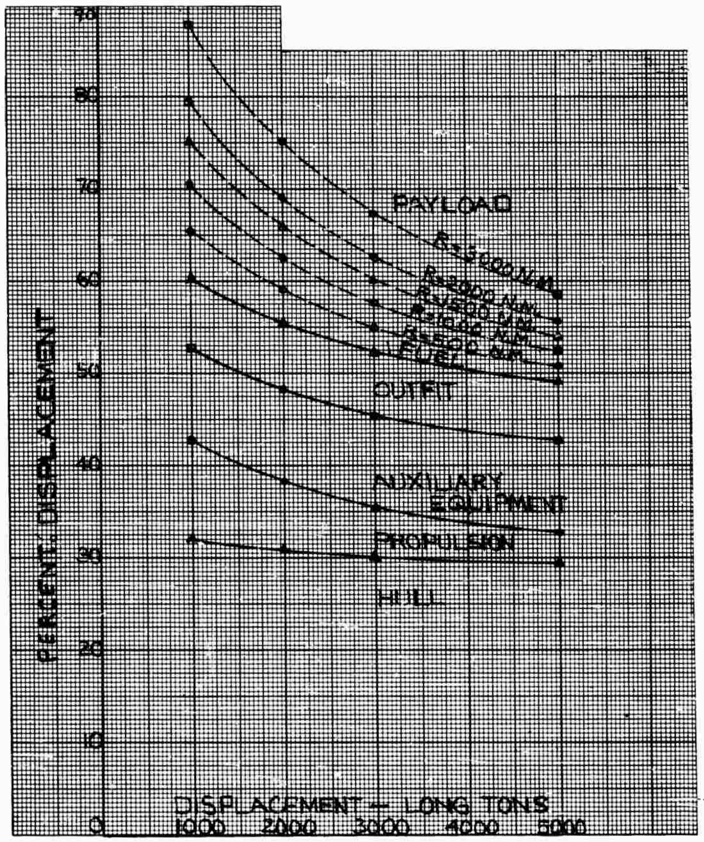


FIG. A-30 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 30 Knots; Various Ranges)

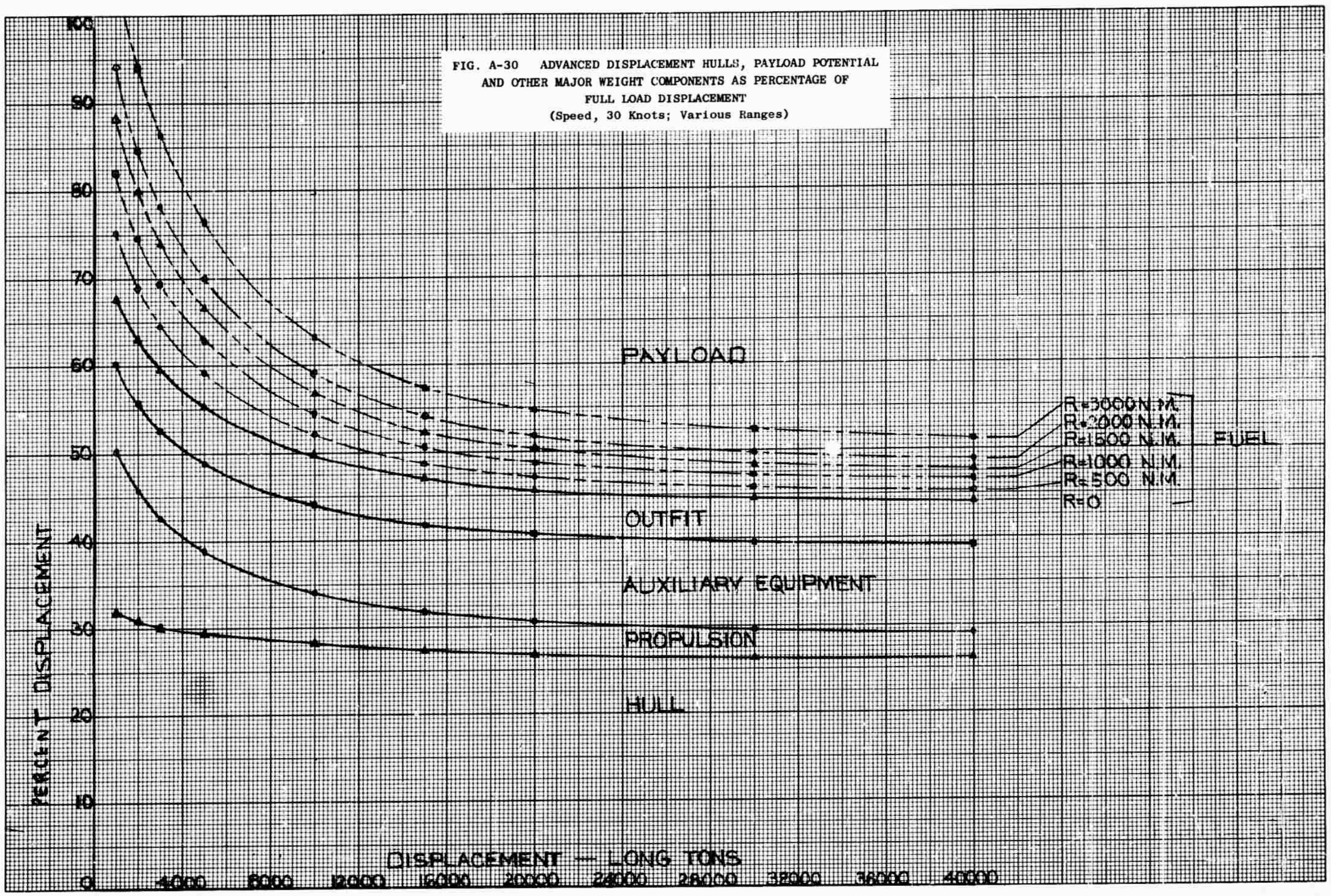


FIG. A-31 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL
AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF
FULL LOAD DISPLACEMENT
(Speed, 40 Knots; Various Ranges)

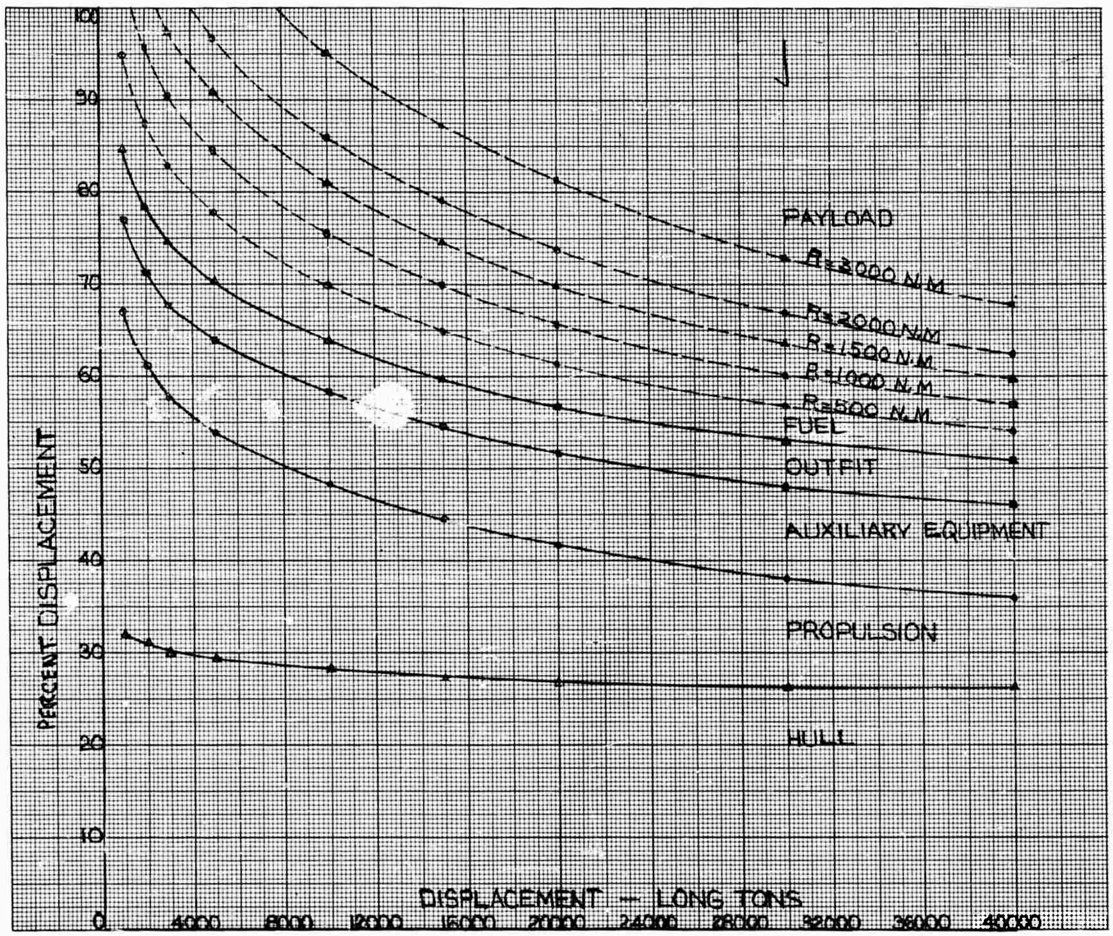


FIG. A-32 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 50 Knots; Various Ranges)

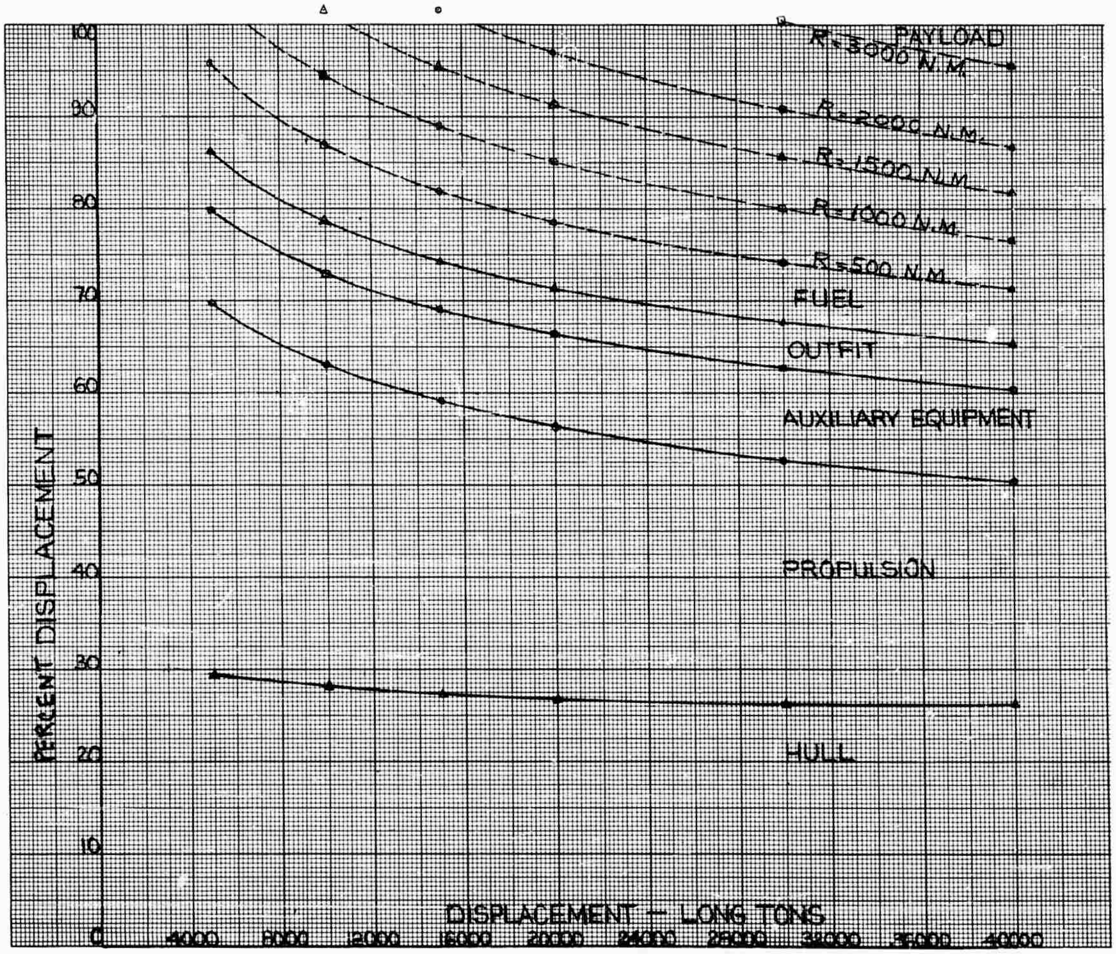


FIG. A-33 ADVANCED DISPLACEMENT
HULLS, PAYLOAD POTENTIAL AND
OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD
DISPLACEMENT
(Speed, 55 Knots; Various Ranges)

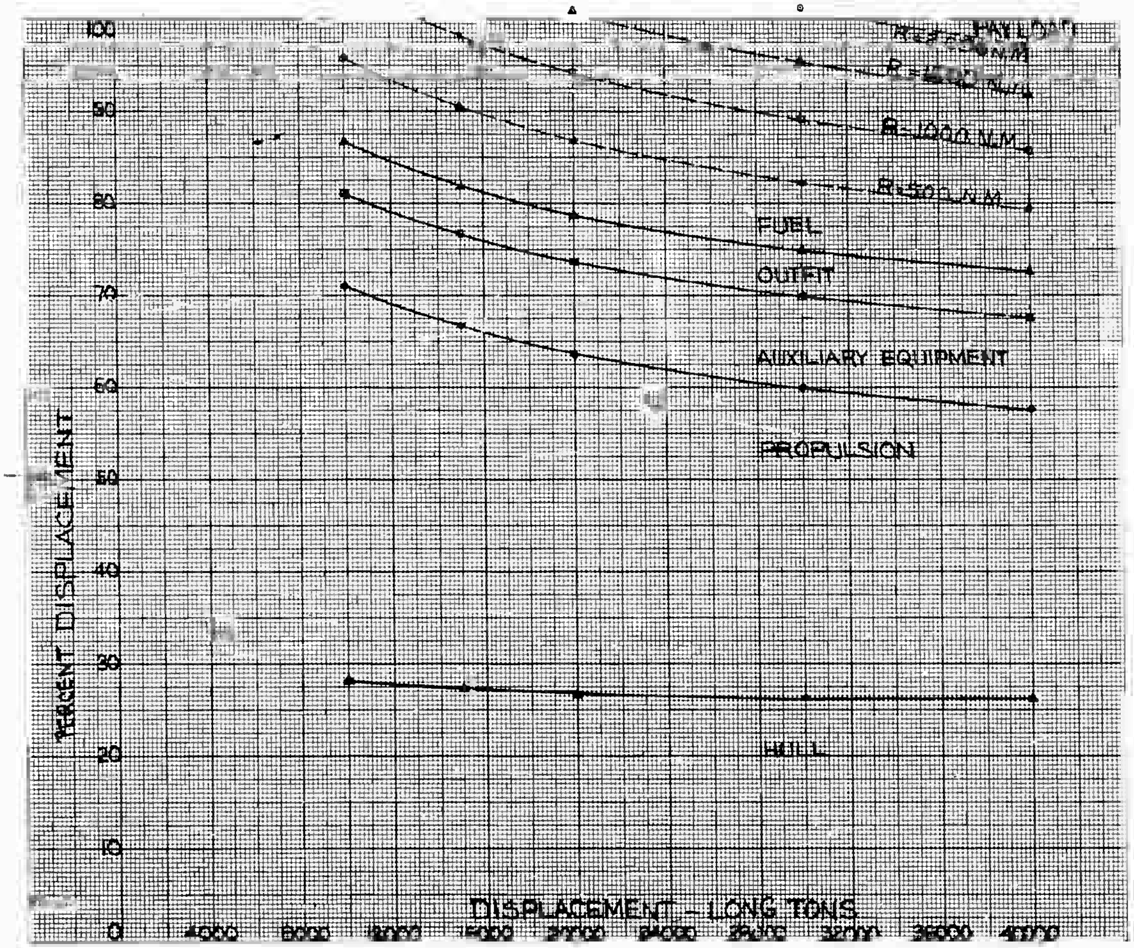


FIG. A-34 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 60 Knots: Various Ranges)

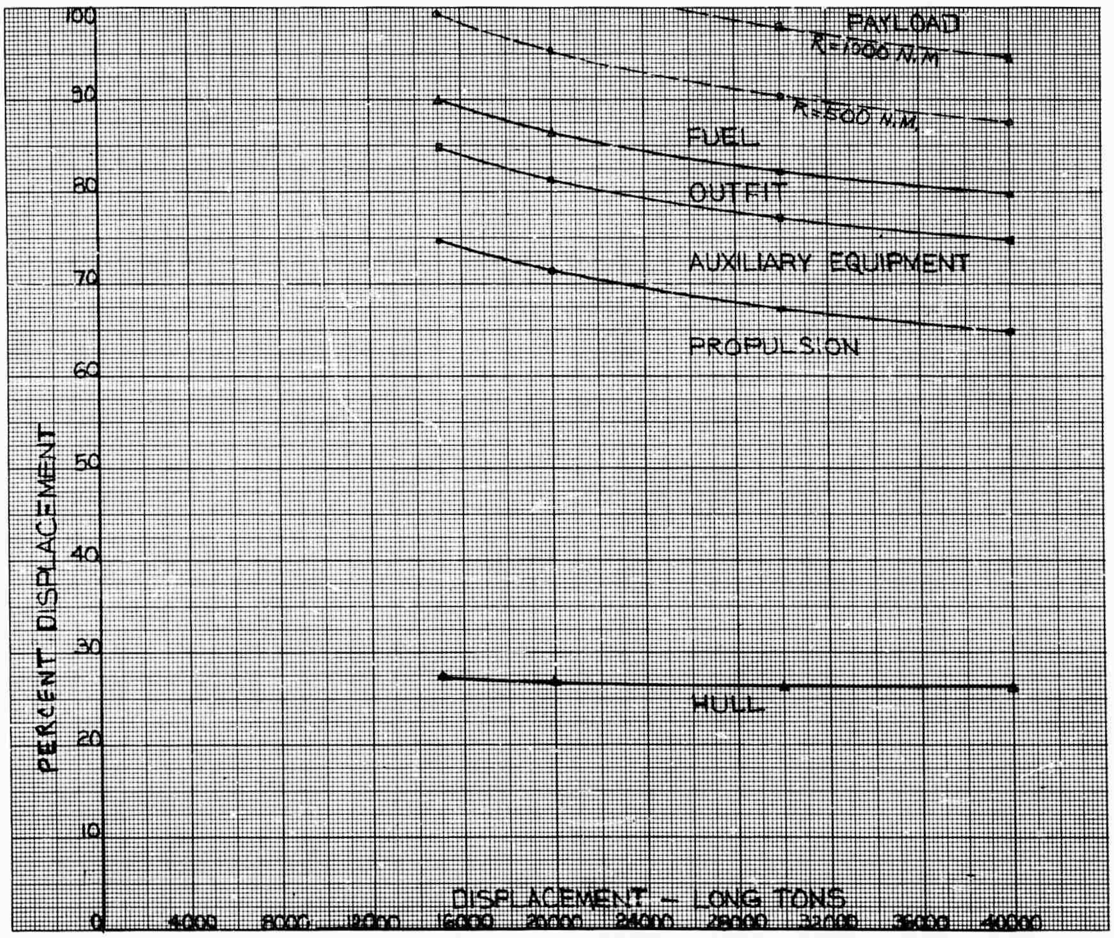


FIG. A-35 ADVANCED DISPLACEMENT HULLS, PAYLOAD VERSUS RANGE FOR DISPLACEMENTS OF 1,000 TO 5,000 TONS
(Various Speeds)

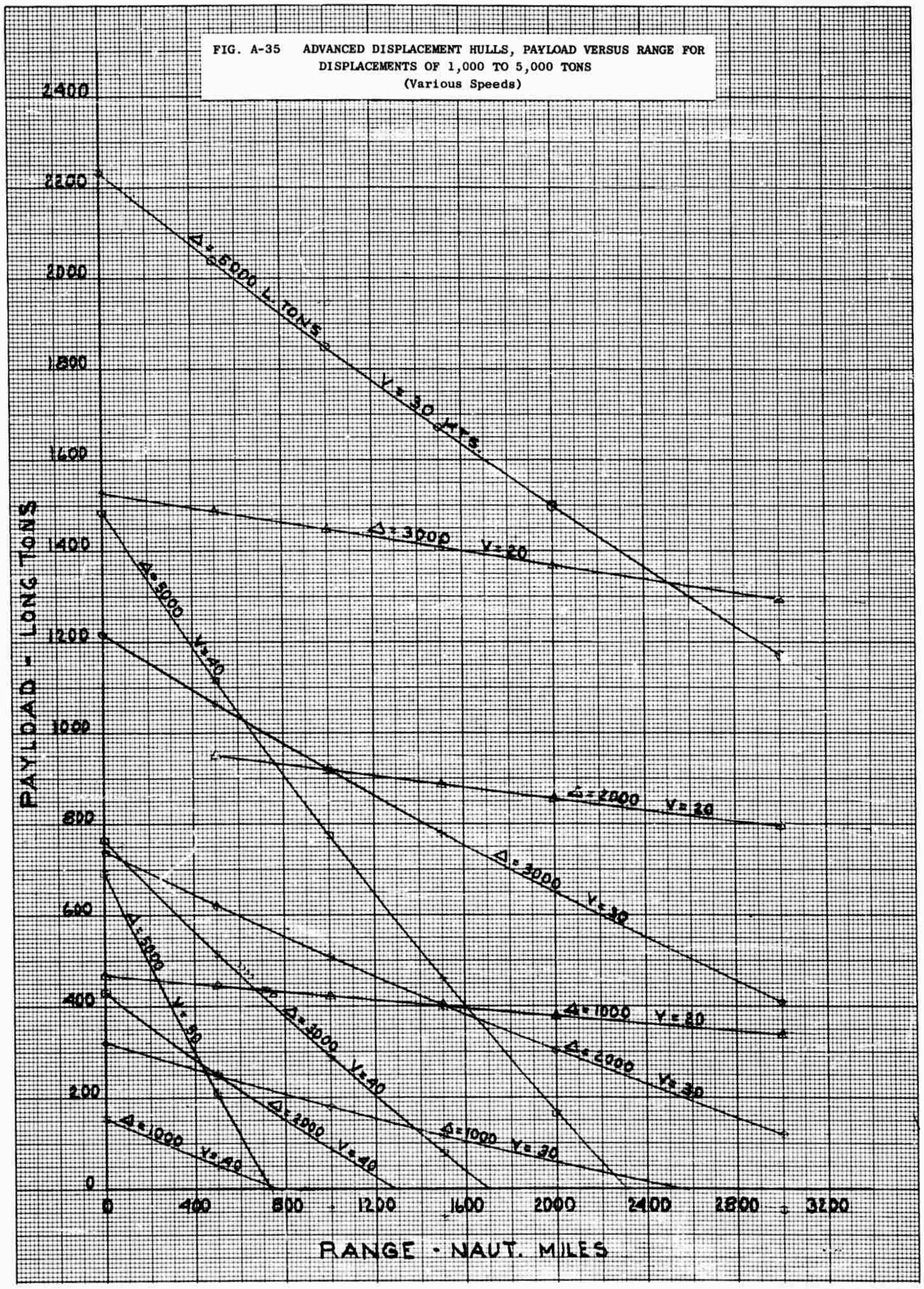


FIG. A-36 ADVANCED DISPLACEMENT HULLS, PAYLOAD VERSUS RANGE FOR DISPLACEMENTS OF 10,000 TO 40,000 TONS (Various Speeds)

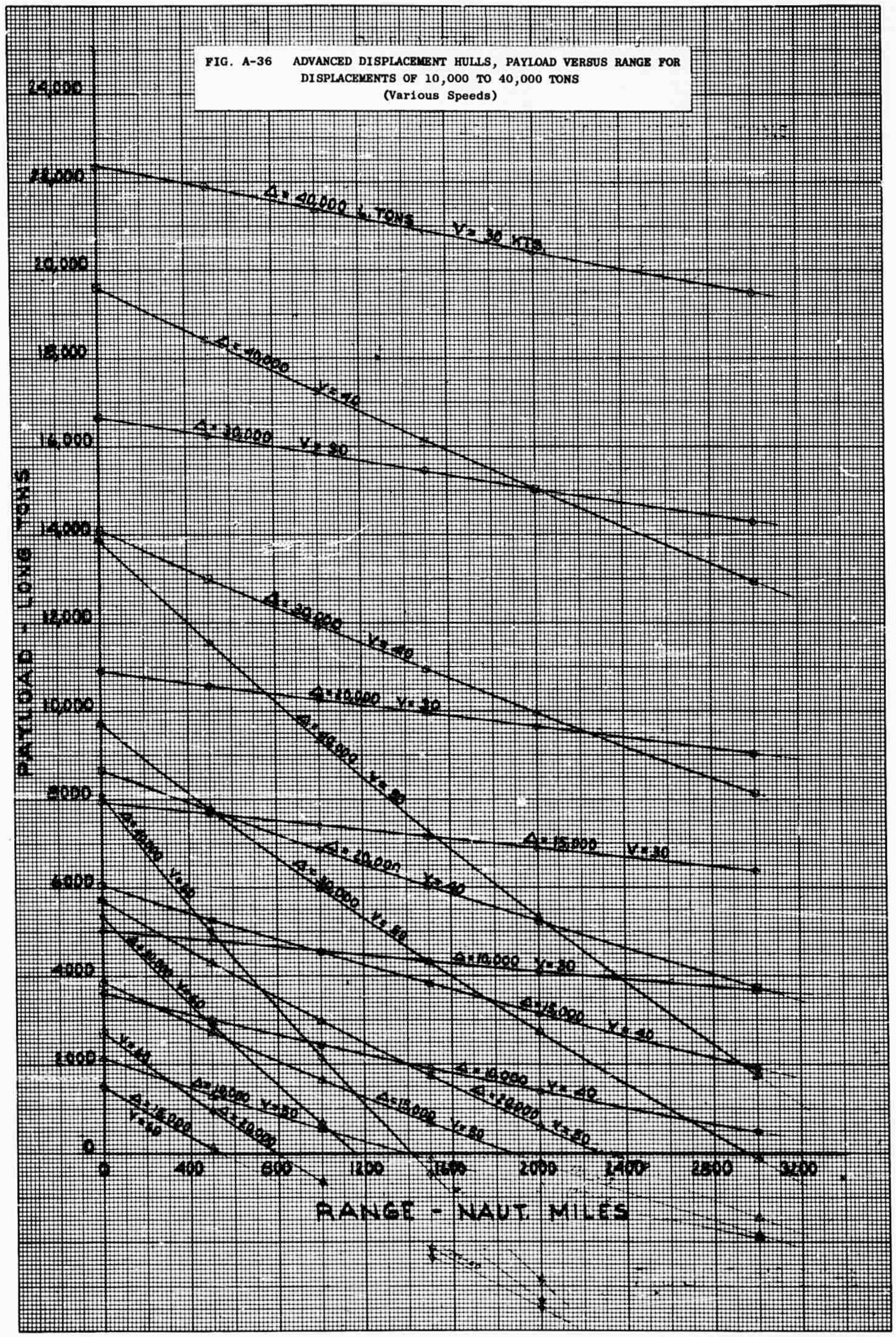


FIG. A-37 ADVANCED DISPLACEMENT HULLS, LENGTH, BEAM, AND DRAFT AS A FUNCTION OF TOTAL DISPLACEMENT

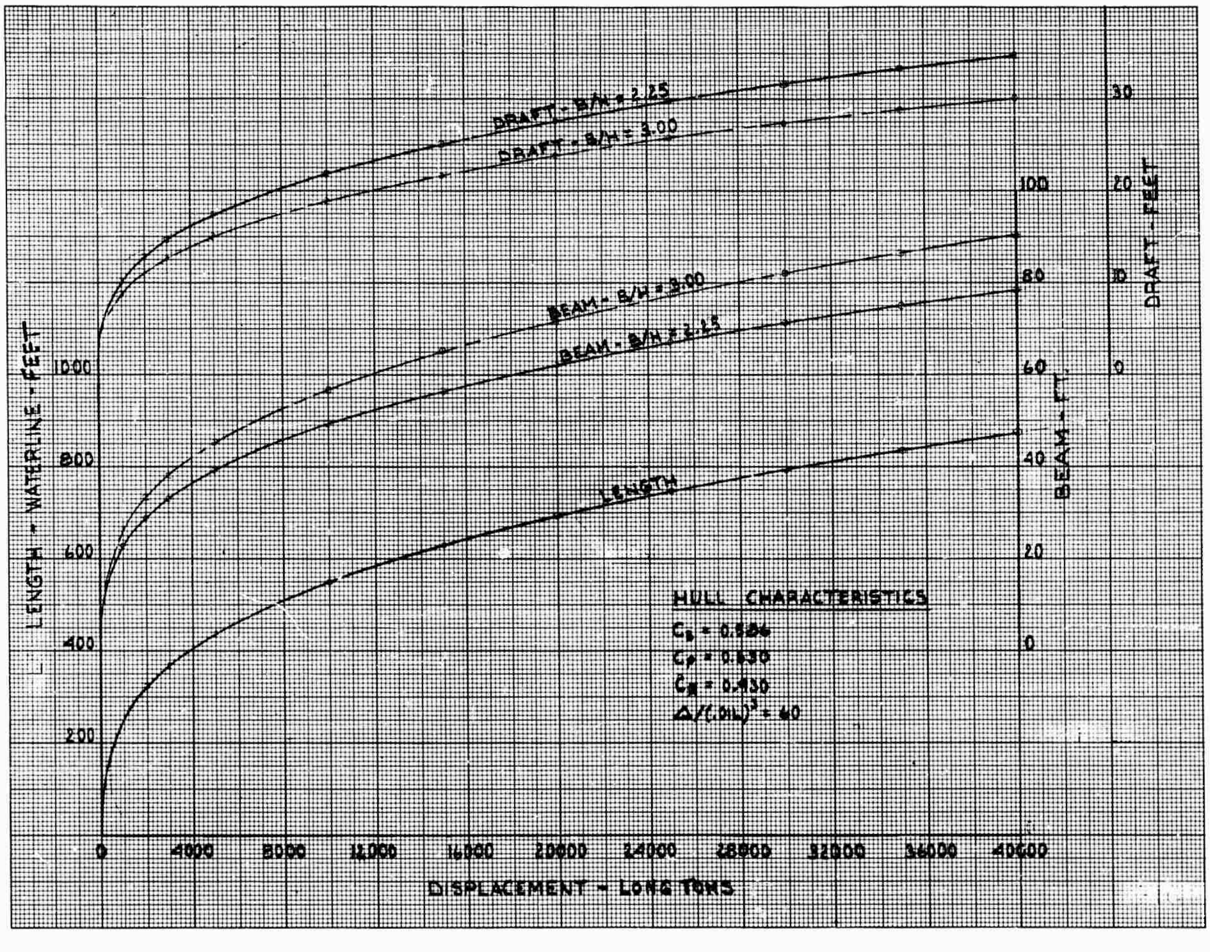


FIG. A-38 ADVANCED DISPLACEMENT HULLS, WEIGHT PER SHAFT HORSEPOWER VERSUS SHAFT HORSEPOWER, LIGHTWEIGHT NUCLEAR POWER PLANT
(See Reference 7)

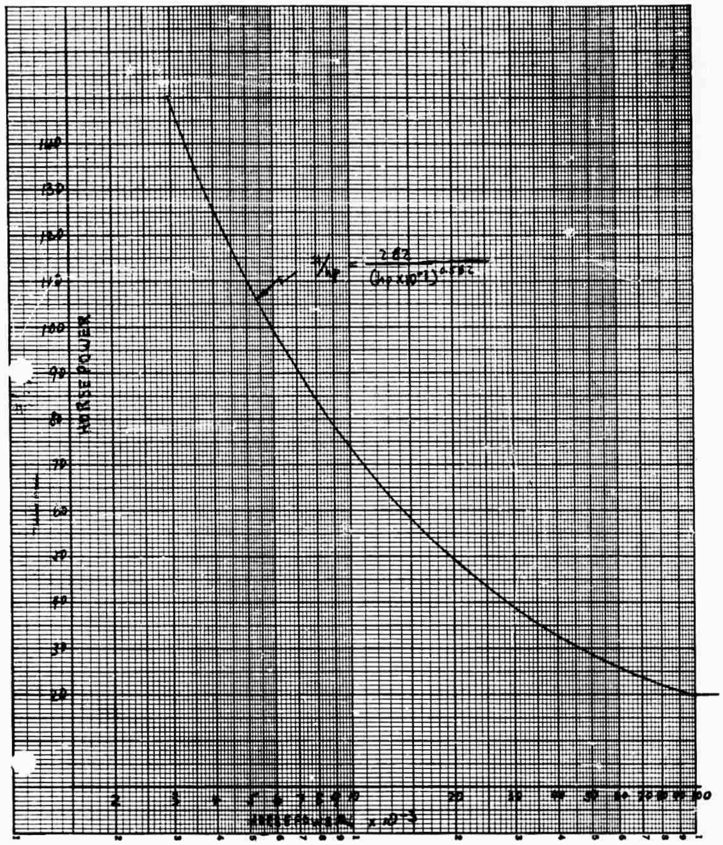


FIG. A-39 ADVANCED DISPLACEMENT HULLS, PAYLOAD AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
NUCLEAR POWER PLANT
(Displacement, Up to 5,000 Tons)

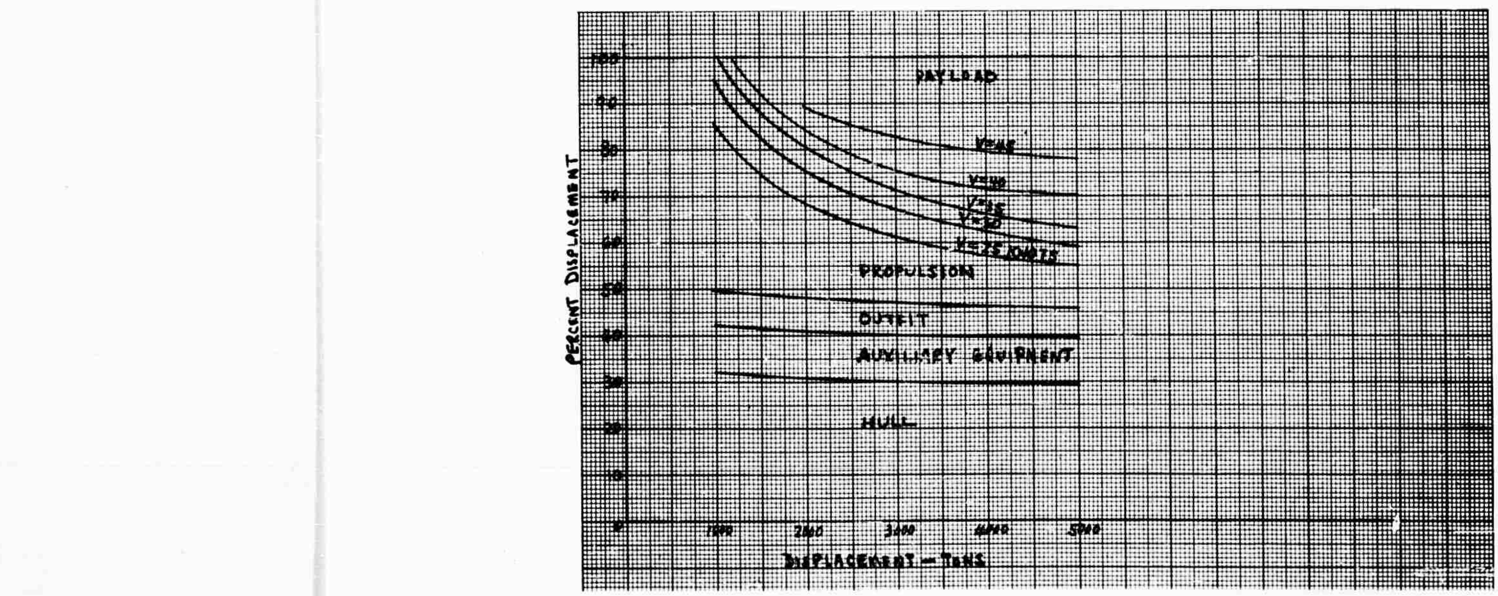
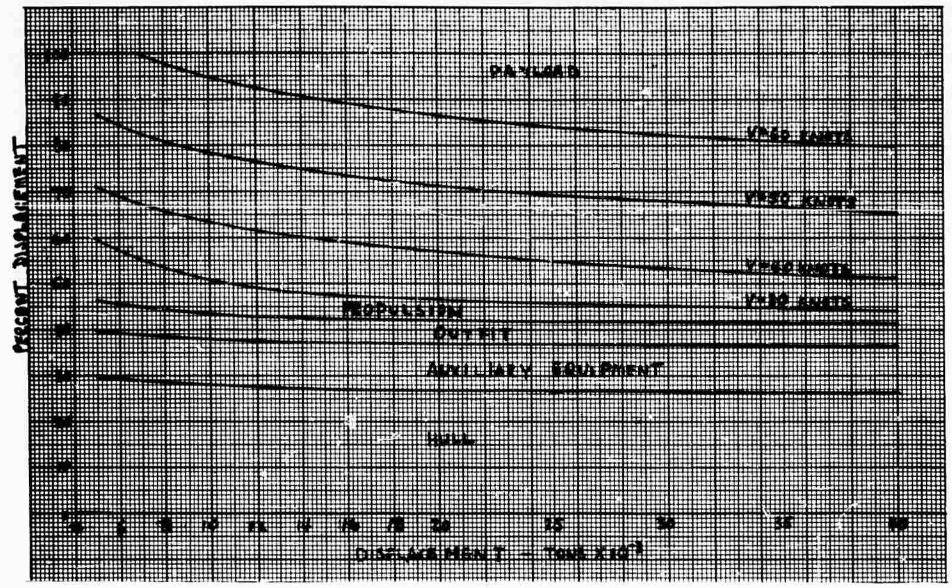


FIG. A-40 ADVANCED DISPLACEMENT HULLS, PAYLOAD AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
NUCLEAR POWER PLANT
(Displacement, 5,000 to 40,000 Tons)



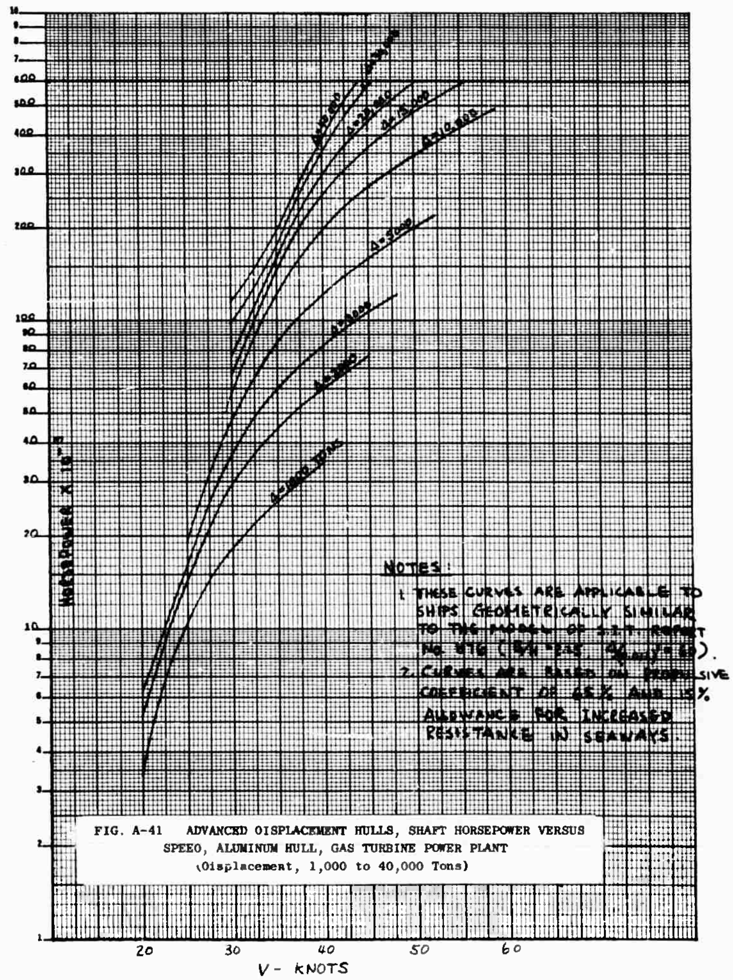


FIG. A-41 ADVANCED DISPLACEMENT HULLS, SHAFT HORSEPOWER VERSUS SPEED, ALUMINUM HULL, GAS TURBINE POWER PLANT
(Displacement, 1,000 to 40,000 Tons)

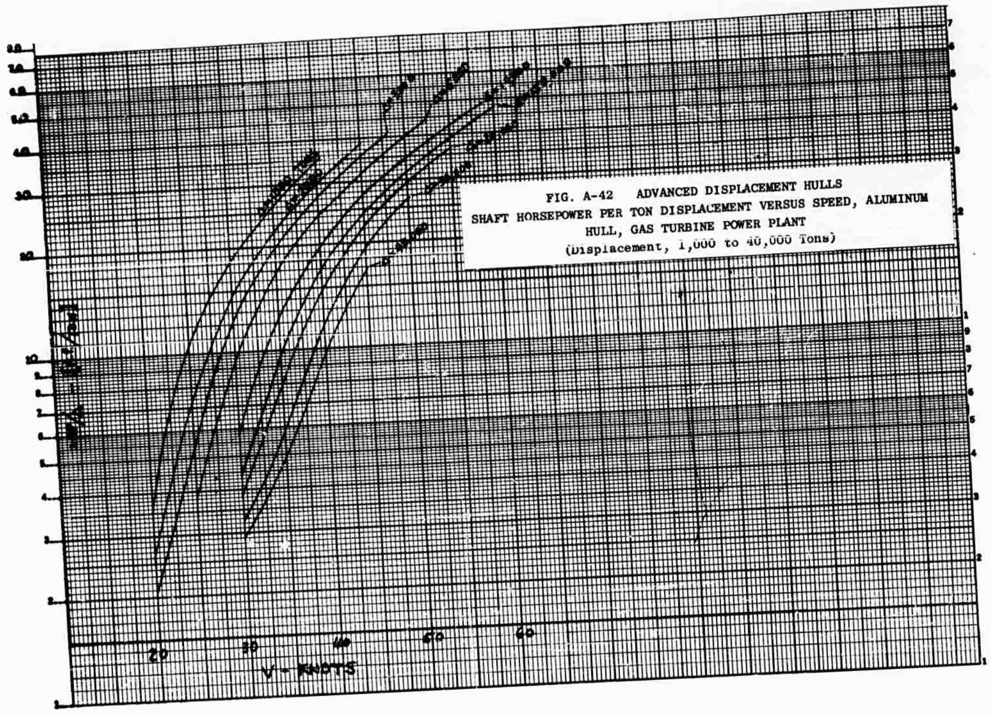
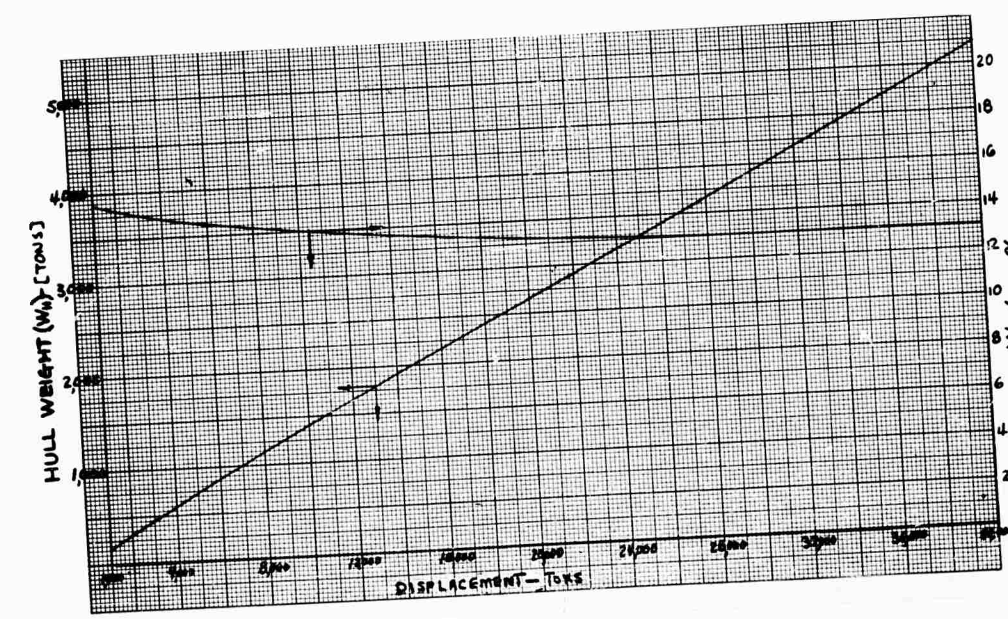


FIG. A-42 ADVANCED DISPLACEMENT HULLS
SHAFT HORSEPOWER PER TON DISPLACEMENT VERSUS SPEED, ALUMINUM HULL, GAS TURBINE POWER PLANT
(Displacement, 1,000 to 40,000 Tons)

FIG. A-43 ADVANCED DISPLACEMENT HULLS
WEIGHT VERSUS FULL LOAD DISPLACEMENT, ALUMINUM HULL



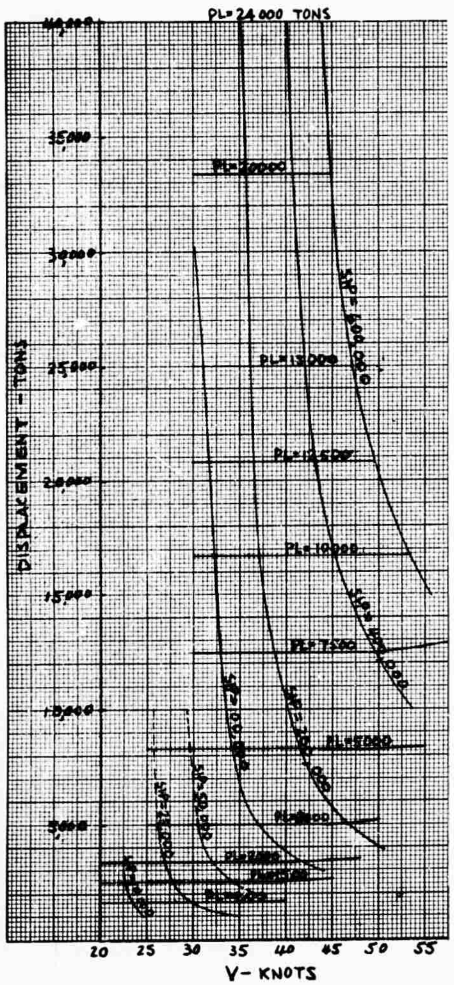


FIG. A-44 ADVANCED DISPLACEMENT HULLS DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER ALUMINUM HULL, GAS TURBINE POWER PLANT (Range, 500 Nautical Miles)

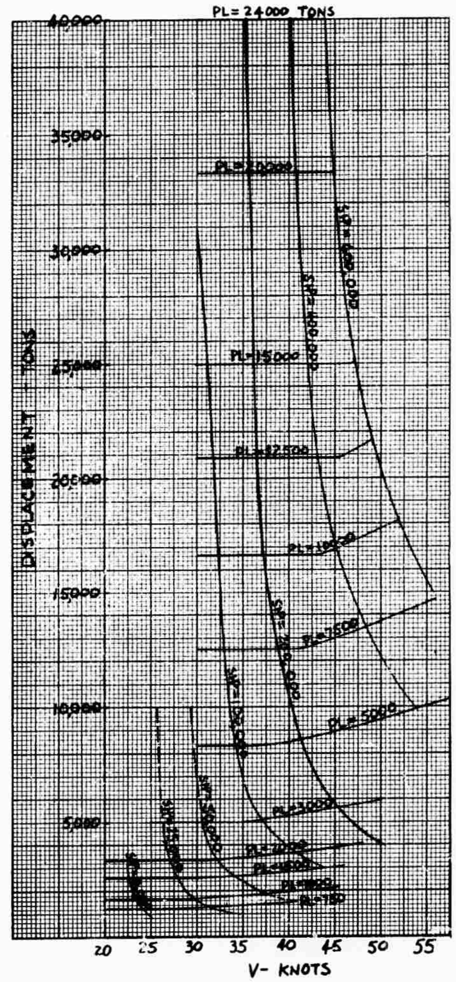


FIG. A-45 ADVANCED DISPLACEMENT HULLS DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER ALUMINUM HULL, GAS TURBINE POWER PLANT (Range, 1,000 Nautical Miles)

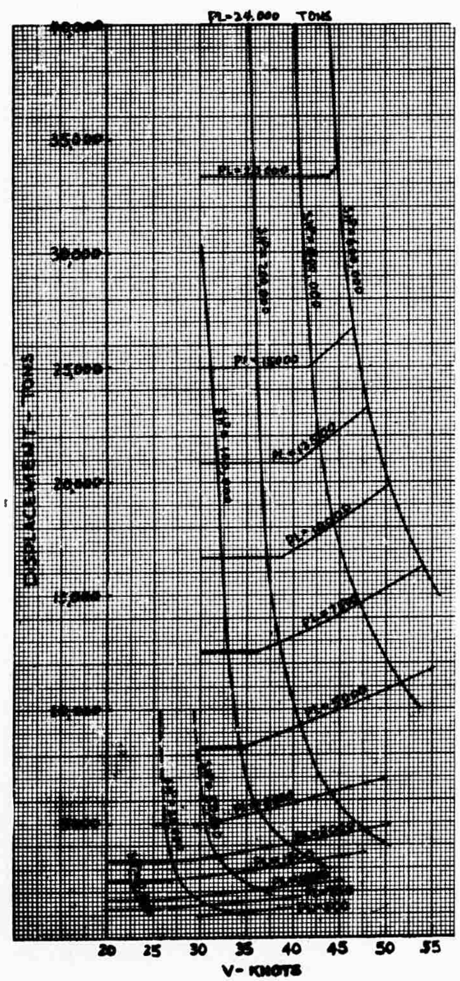


FIG. A-46 ADVANCED DISPLACEMENT HULLS DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER ALUMINUM HULL, GAS TURBINE POWER PLANT (Range, 1,500 Nautical Miles)

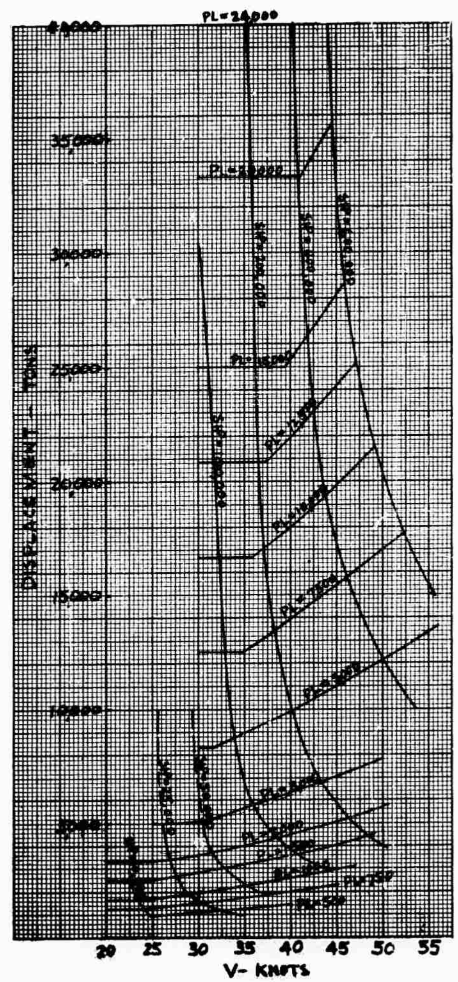


FIG. A-47 ADVANCED DISPLACEMENT HULLS DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER ALUMINUM HULL, GAS TURBINE POWER PLANT (Range, 2,000 Nautical Miles)

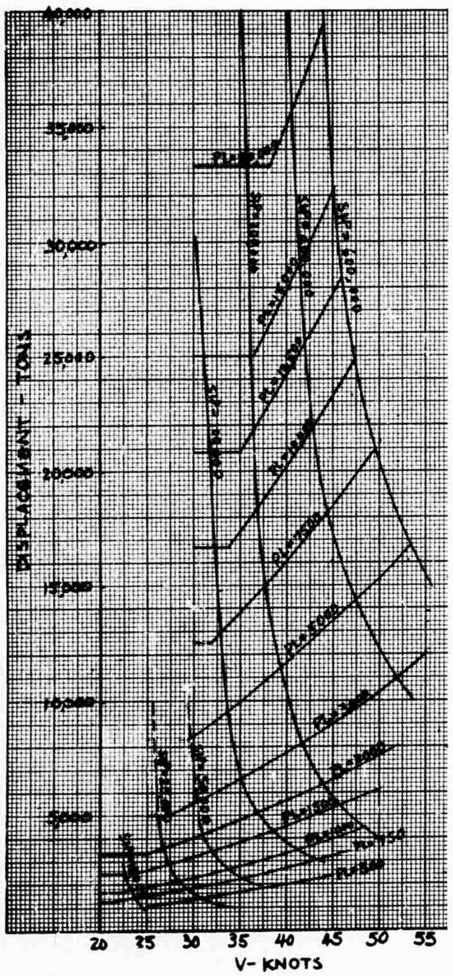


FIG. A-48 ADVANCED DISPLACEMENT HULLS
DISPLACEMENT AND PAYLOAD VERSUS SPEED
AND REQUIRED SHAFT HORSEPOWER
ALUMINUM HULL, GAS TURBINE POWER PLANT
(Range, 3,000 Nautical Miles)

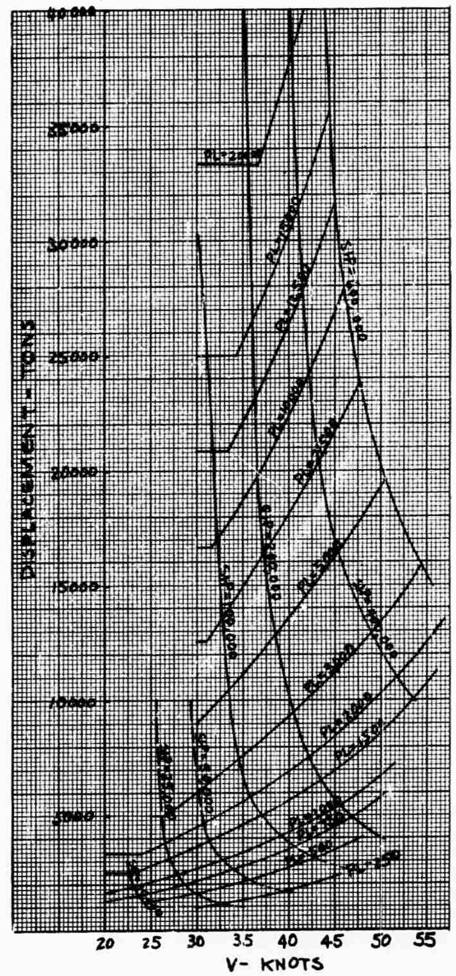


FIG. A-49 ADVANCED DISPLACEMENT HULLS
DISPLACEMENT AND PAYLOAD VERSUS SPEED
AND REQUIRED SHAFT HORSEPOWER
ALUMINUM HULL, GAS TURBINE POWER PLANT
(Range, 4,000 Nautical Miles)

FIG. A-50 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
ALUMINUM HULL, GAS TURBINE POWER PLANT
(Speed, 25 Knots; Various Ranges)

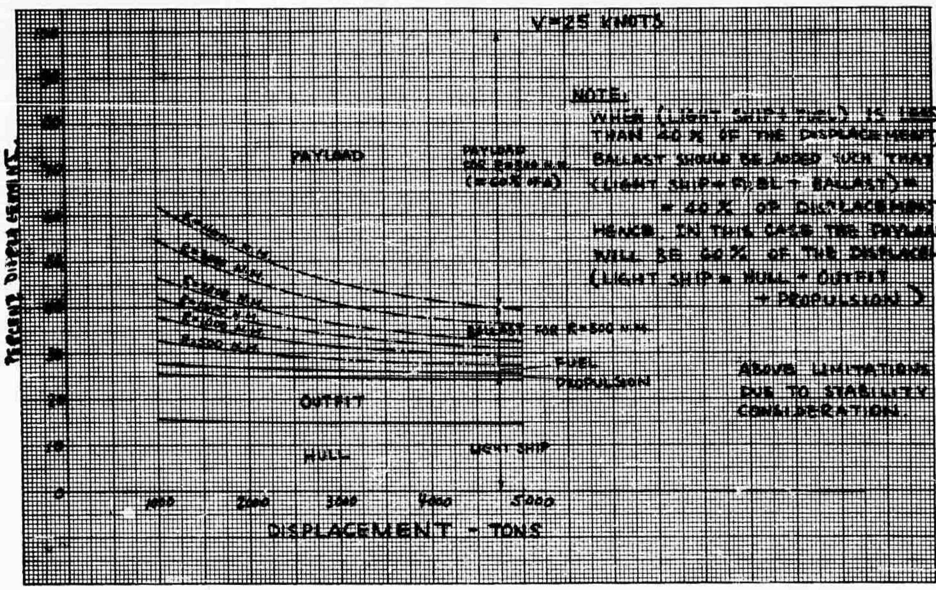


FIG. A-51 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
ALUMINUM HULL, GAS TURBINE POWER PLANT
(Speed, 30 Knots; Various Ranges)

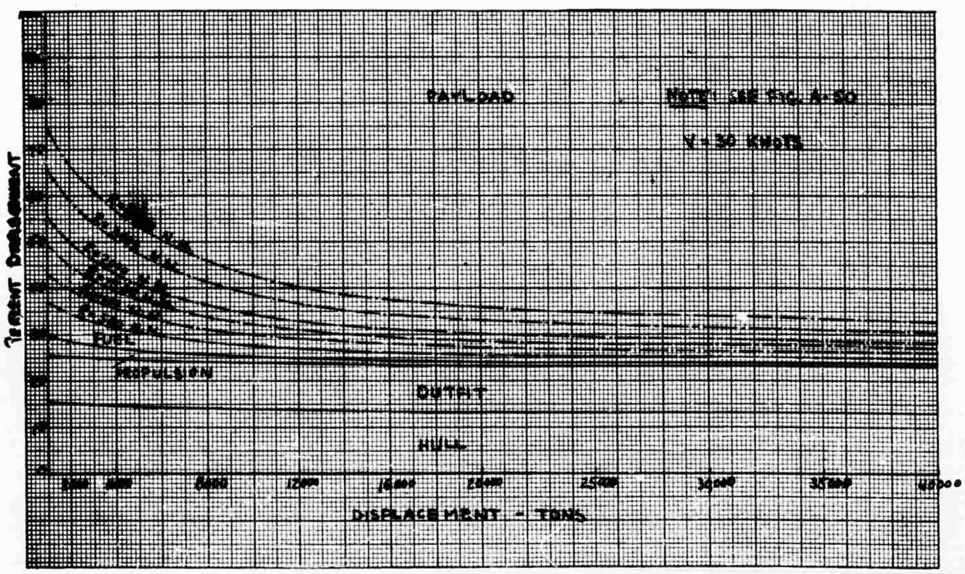


FIG. A-52 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT ALUMINUM HULL, GAS TURBINE POWER PLANT (Speed, 35 Knots; Various Ranges)

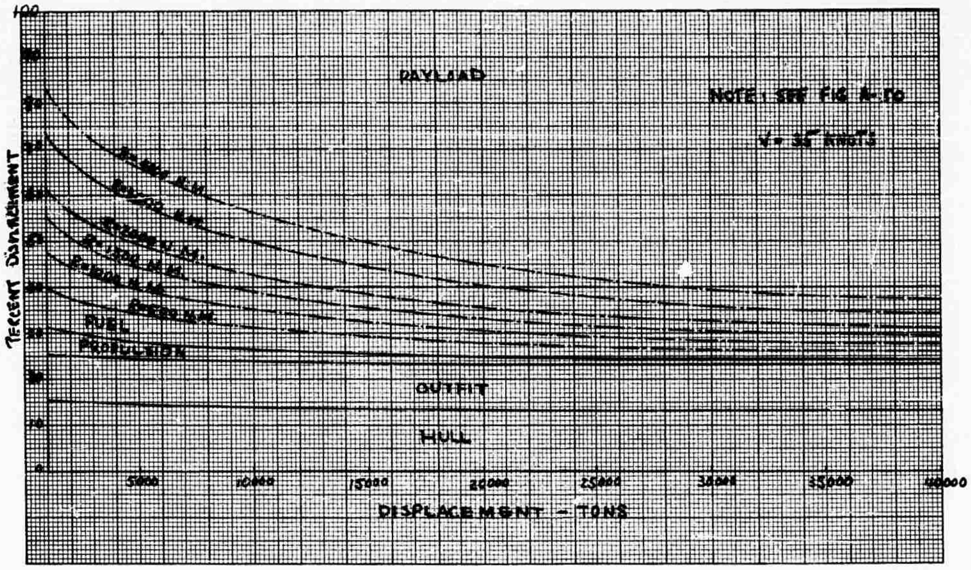


FIG. A-53 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIONT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT ALUMINUM HULL, GAS TURBINE POWER PLANT (Speed, 40 Knots; Various Ranges)

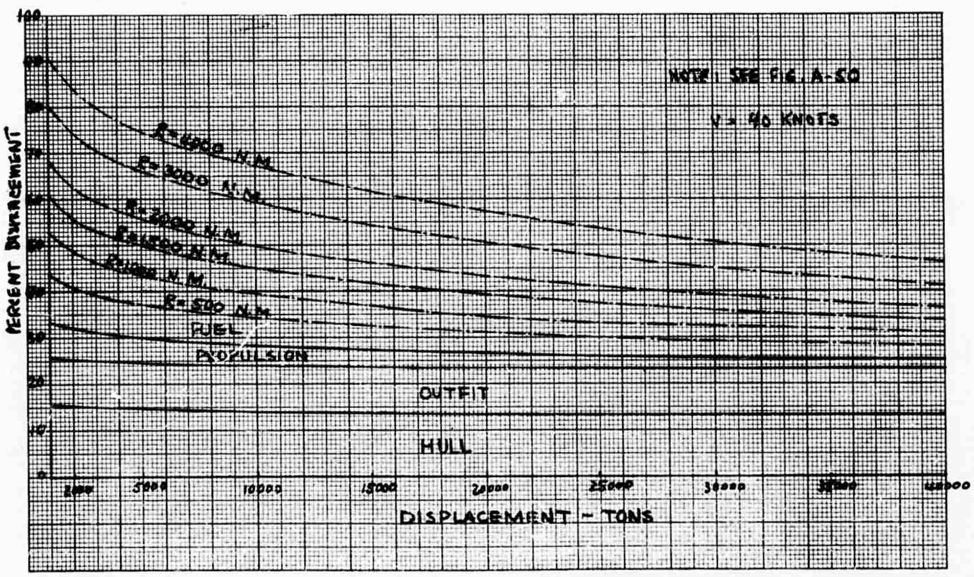


FIG. A-54 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT ALUMINUM HULL, GAS TURBINE POWER PLANT (Speed, 45 Knots; Various Ranges)

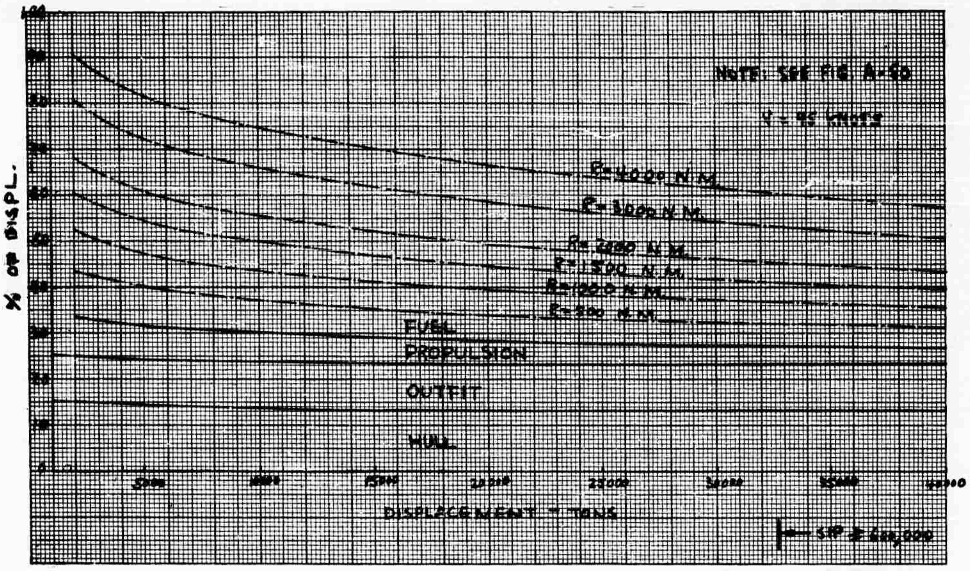


FIG. A-55 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT ALUMINUM HULL, GAS TURBINE POWER PLANT (Speed, 50 Knots; Various Ranges)

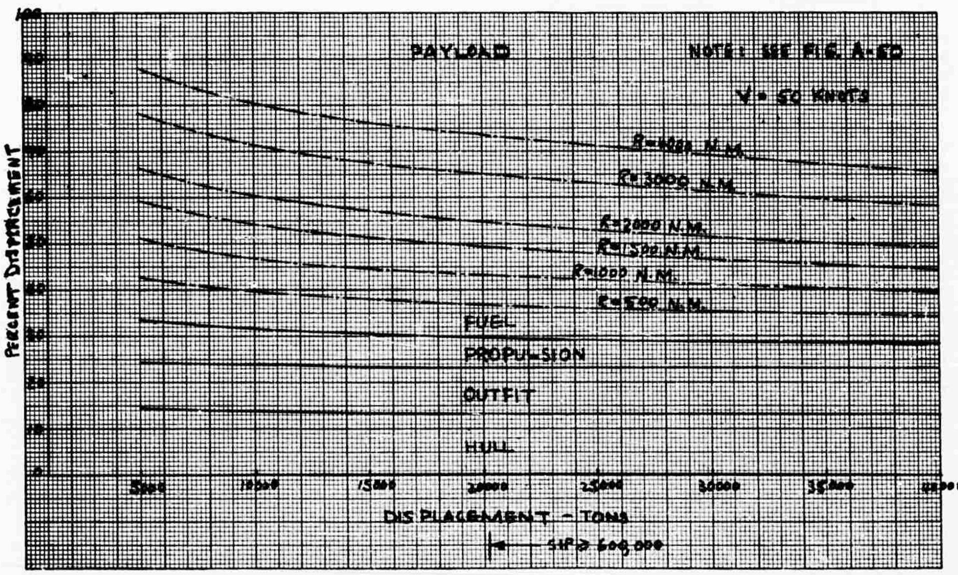


FIG. A-56 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
ALUMINUM HULL, GAS TURBINE POWER PLANT
(Speed, 55 Knots; Various Ranges)

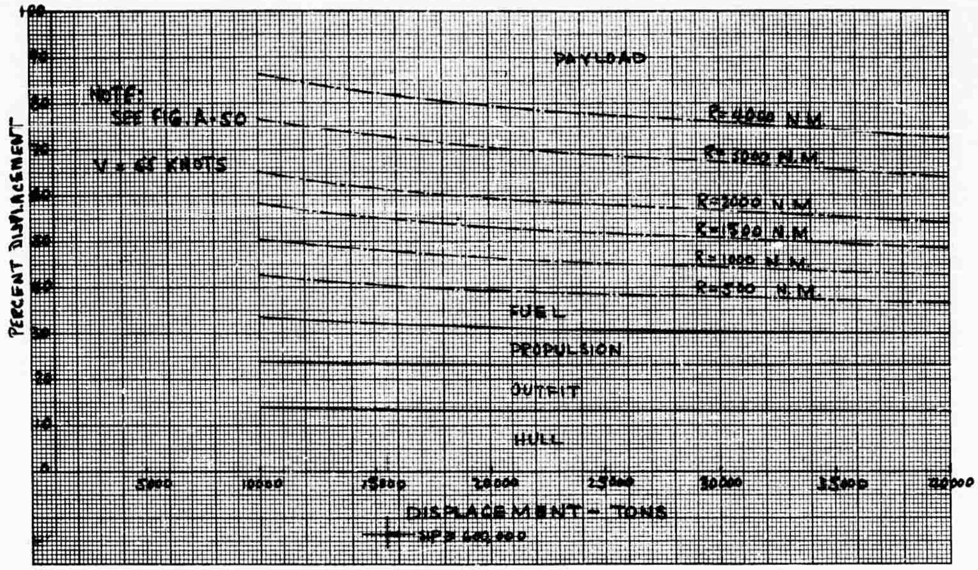


FIG. A-57 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
ALUMINUM HULL, NUCLEAR POWER PLANT
(Displacement, 1,000 to 5,000 Tons; Various Speeds)

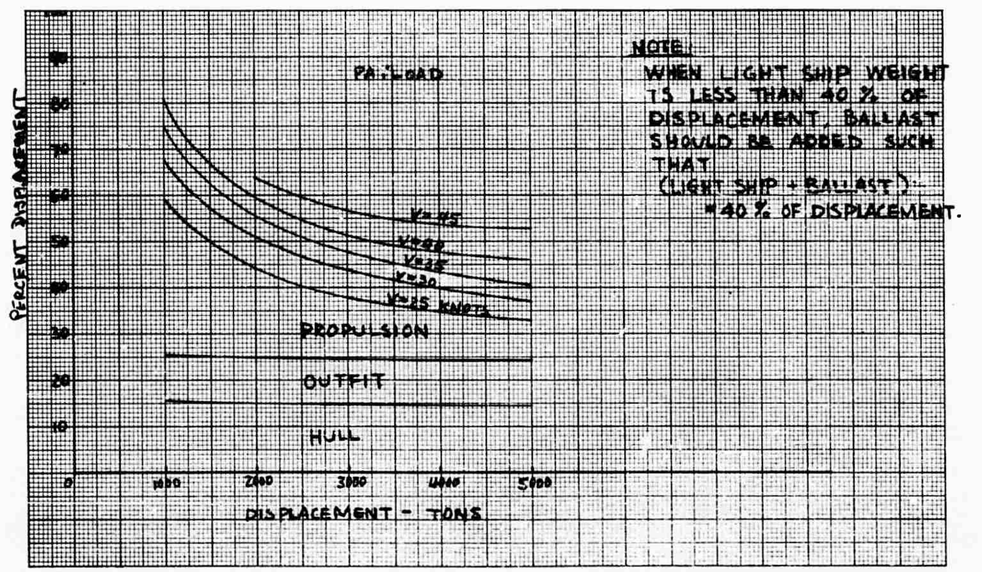


FIG. A-58 ADVANCED DISPLACEMENT HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT
ALUMINUM HULL, NUCLEAR POWER PLANT
(Displacement, 5,000 to 40,000 Tons; Various Speeds)

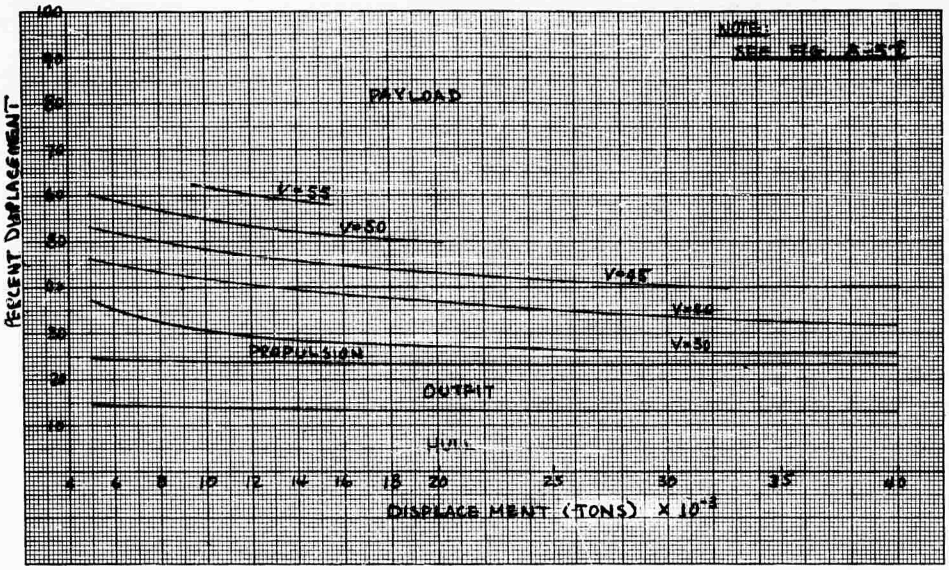


FIG. A-59 ADVANCED DISPLACEMENT HULLS, PAYLOAD VERSUS RANGE
ALUMINUM HULL, GAS TURBINE POWER PLANT
(Displacement, 1,000 to 5,000 Tons; Range, 500 to 4,000 Nautical Miles)

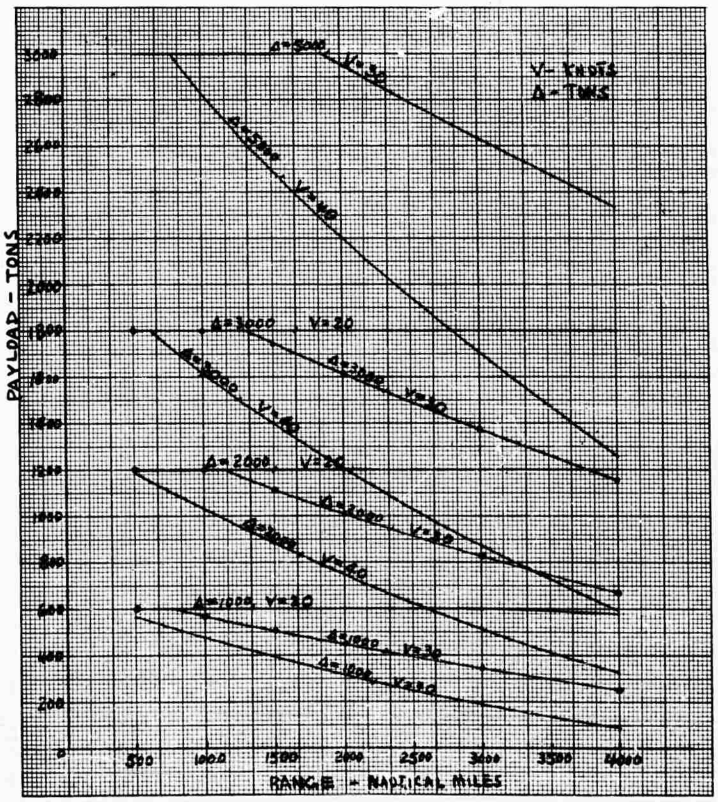
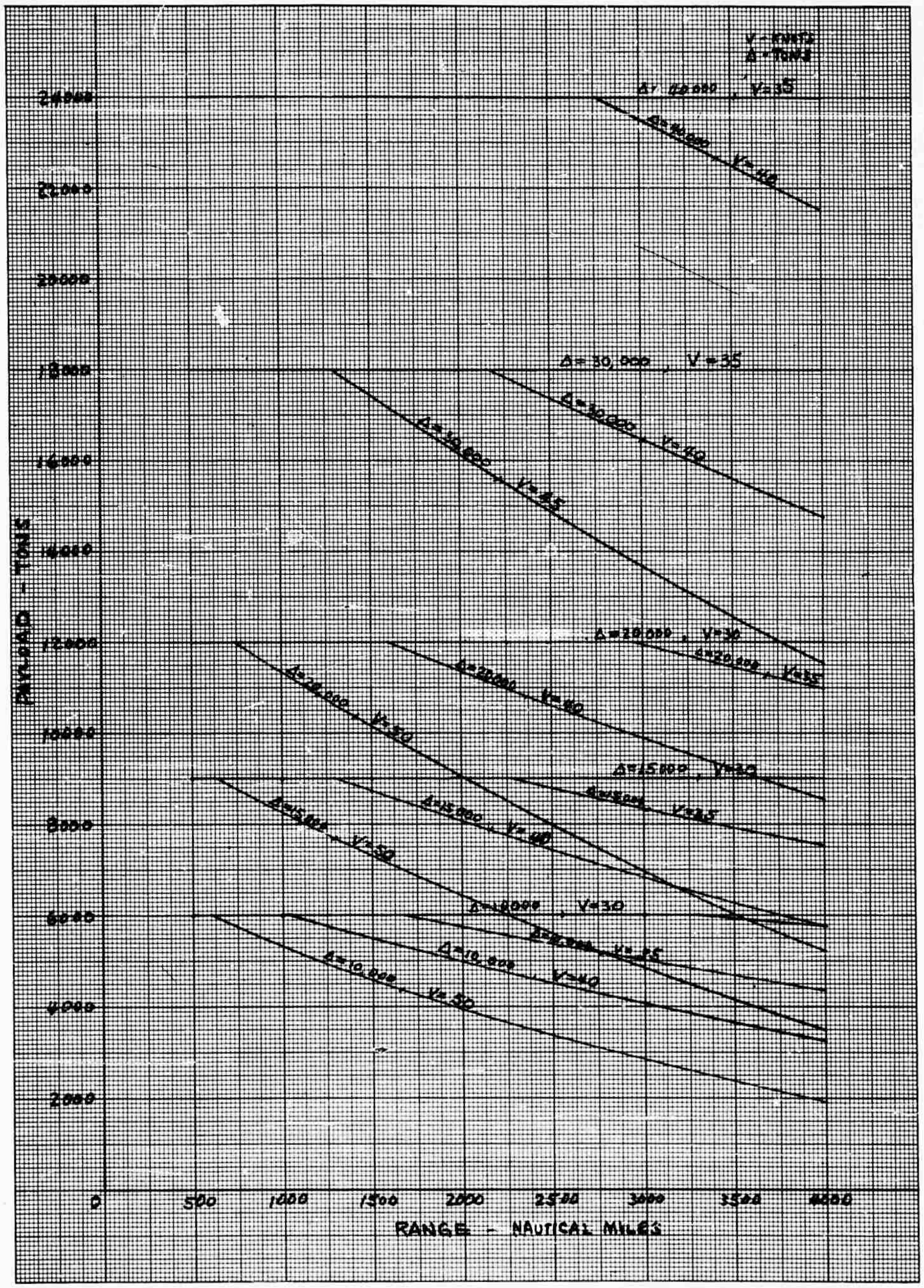


FIG. A-60 ADVANCED DISPLACEMENT HULLS, PAYLOAD VERSUS RANGE
ALUMINUM HULL, GAS TURBINE POWER PLANT
(Displacement, 10,000 to 40,000 Tons; Range, 500 to 4,000 Nautical Miles)



Appendix B

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Appendix B

PLANING HULLS

Appendix B

PLANING HULLS

Introduction

The purpose of this study is to provide information for assessing the feasibility of planing hulls for amphibious fleet units in the 1970-1980 time period. Thus far the planing hull has been employed primarily for high-speed boats up to approximately 100,000 pounds displacement. The objections to extending its use to larger sizes seem to be:

1. The lowest speed at which the planing hull becomes advantageous in reduced resistance, as compared to hull forms such as those for destroyers, increases with displacement. The operation of large draft boats at high speed requires great power, with its associated space problem and with the even greater problem of converting the power to thrust.[1]

2. Poor seaworthiness of planing hulls (as compared to displacement hull forms and hydrofoils).

The power problem may be relieved somewhat by the development of lightweight power plants such as aircraft-type gas turbines. The seaworthiness of the planing hull is discussed in a later section.

An estimate of the power required and payload available for planing hulls of displacements up to 3,000 tons and speeds up to 100 knots will be considered below with the assumption that a high-powered, lightweight gas turbine plant of 3 lbs/HP will be available in the time period considered.

Brief consideration is also given to the use of nuclear power as a means of improving performance.

Resistance Estimates

A number of towing test results of planing hull models have been published by DTMB.[2-5] The assumed prototype of these models weighed approximately 100,000 pounds. According to the laws of similitude, however, each model can represent a hull of any size. A semiempirical

calculation method for predicting the still-water performance of planing hulls is presented by Clement and Pope.[6,7] Examples were given by these authors that show good agreement between predicted resistance by calculation, and model tests, in the planing region. In the region where F_∇ is smaller than approximately 3.5,* an appreciable portion of the load is supported by buoyancy and the calculated resistance is not valid.[7]

The above-mentioned calculation method was used in this study to estimate the performance of stepless planing hulls in the planing region. The maximum beam over spray strips, b, selected for calculation is presented in Fig. B-1.

The average deadrise chosen was 10° and lcp/b = 1.5 was assumed. A roughness allowance of 0.004 and 5 percent for air drag were also assumed. The results are presented in Fig. B-2 as solid lines. The dotted line in the figure was drawn by referring to the model test data of Reference No. 4 to represent the resistance characteristics of planing hulls in the semiplaning region. This may seem rather arbitrary, since the hull forms are not the same and one line cannot represent all sizes. However, it is believed to be adequate for estimating purposes.

Calculation of Shaft Horsepower and Fuel Requirements

Given values of speed and displacement, the corresponding Froude number is computed as follows:

$$F_\nabla = \frac{1.689V}{\sqrt{g\nabla^{1/3}}}$$

V = speed (knots)

∇ = volumetric displacement (ft^3).

Knowing F_∇ and R_T/Δ, the total specific resistance is obtained from Fig. B-2. EHP is then obtained as follows:

$$EHP = 1.689V \times \frac{R_T}{\Delta} \times \Delta \times \frac{2240}{550} = 6.88\ V\ \frac{R_T}{\Delta}\ \Delta \quad .$$

* Notations in this study are those of Reference Nos. 6 and 7.

To obtain SHP, a propulsive coefficient must be assumed. In this study, a value of 0.55 was used, which seems to be a reasonable value for a highly loaded propeller(s) on a high-speed craft. An allowance of 15 percent for seaway operation was also assumed. Thus,

$$SHP = \frac{1.15\ EHP}{0.55} = 14.4 \left(\frac{R_T}{\Delta}\right) V \Delta \quad .$$

Fig. B-3 shows SHP as a function of speed for a range of displacements. The solid line portion shows the true planing operation and the dotted line portion shows the semiplaning operation.

Specific fuel weight is calculated using Breguet's range equation, which accounts for the reduction in fuel consumption as the weight of remaining fuel, and hence displacement is reduced.

$$\frac{W_F}{\Delta} = 1 - e^{-x}$$

where

$$x = \frac{R \times SFC \left(\frac{R_T}{\Delta} \times 1.15\right)}{325\ N_{pc}} = 0.00645 \times R \times SFC \times \left(\frac{R_T}{\Delta}\right) \quad .$$

This applies to the fuel powered engine only; the nuclear power plant has no fuel weight.

A specific fuel consumption for an open cycle, aircraft-type gas turbine marine engine of 0.55 lb/SHP/hour was used. This figure should be attainable in very large power levels during the time period anticipated.

Nuclear Power Plant

A study of the effect on the potentialities of planing hulls of a gas-cooled nuclear reactor driving a geared gas turbine was made. Values of payload as a percent of displacement were computed for a range of speeds and displacements. Weight of such a power plant was based on Reference No. 8 (refer also to Hydrofoil study, Appendix C). Range is no longer a variable since there is no real fuel consumption. The results are shown on Fig. B-4.

Weight and Payload

It is possible to equate range, payload, and speed, which are the three parameters related to a specific mission, in one expression as follows:

$$\Delta = LS + PL + W_F \tag{1}$$

where

LS = light ship weight (long ton)

PL = payload (long ton)

W_F = weight of fuel (long ton)

$$LS = W_h + W_m + W_o = \text{light ship weight (long ton)} \tag{2}$$

where

W_h = 0.25Δ = hull weight (long ton)

W_o = 0.10Δ = outfit weight (long ton)

W_m = $3\ [\text{lbs/SHP}] \times \text{SHP} \times \frac{1}{2240}$ = machinery weight (long ton).

Figures B-5 through B-14 show the relationship between weights, speeds, and ranges.

Figure B-15 shows the effect of varying payload and range for vehicles of constant speed and displacement.

Twenty-five percent and 10 percent of displacements as factors to allow for hull weight and outfit weight are based on good practice for large displacement steel ships such as tankers. It is assumed that the tendency of the larger dynamic loads to increase the weight would be offset by the use of high strength-to-weight ratio materials such as aluminum.

A power plant weight of 3 lbs/SHP has been assumed.

Hull Configuration

Figure B-1 provides a means of obtaining approximate hull dimensions for any planing hull displacing from 100 to 3,000 long tons. Of course, these curves cannot be considered as design information and should only be used for rough estimation of length, beam, and depth.

It is conceivable that the hydrodynamic performance could be improved, with research and development, to increase the capability of the planing hull beyond the results of this study. Thus far, it has been beyond the scope of this study to predict the performance of planing hulls beyond a direct projection of the present state-of-the-art to larger sizes.

Effect of Sea State on Planing Hull Speed

The effects of surface waves on a planing hull are manifested in several ways, all of which hinder operation. The motion of the hull against irregular, or regular, wave surfaces produces a continuous series of impacts which, first, increase the total resistance and, second, decrease the speed. The amount of speed degradation depends, of course, on the size of the waves. For this reason it is desirable to have a reasonably large increment between minimum planing speed and cruising speed. If the increment is small, an encounter with even small waves will result in alternate planing and displacement operation--an unstable and very inefficient situation.

If the impact forces and their frequency are in certain undesirable, but frequently difficult to avoid, relationships with the virtual mass and natural frequencies of motion of the planing hull, violent motions of the hull, usually pitching and heaving, will result. The impact forces and the induced motions are harmful in that they cause large dynamic structural loads and discomfort and possible danger to the crew. With small planing hulls, it has been found that the wave effects can be reduced by the use of convex bottom sections, particularly near the bow. These sections, however, are not the best for minimizing calm water resistance.

There have been very few scientifically executed tests on small planing hulls in waves and none, to our knowledge, on large hulls. It must be generally true, however, that the maximum wave size in which a planing hull can operate with required speed, stability, safety, and comfort must increase as the size and weight of the vessel increases. The maximum wave size for a design probably increases linearly with a parameter such as displacement to the 1/3 exponent. Thus, if a 100-ton planing hull can

operate satisfactorily in 2-foot waves, the corresponding satisfactory wave height for a 3,000-ton planing hull should be $\left(\frac{3000}{100}\right)^{1/3} \times 2' = 6.2'$. It thus appears that operation in waves will probably put as severe a limitation on planing hulls as on displacement vessels, and undoubtedly will be a greater handicap than to the hydrofoil.

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3. Clement, E.P. and P.M. Kimon, Comparative Resistance Data for Four Planing Boat Designs, DTMB Report 1113, January 1957

4. Clement, E.P., Development and Model Tests of an Efficient Planing Hull Design, DTMB Report 1314, April 1959

5. Clement, E.P. and C.W. Tate, Sr., Smooth Water Resistance of a Number of Planing Boat Designs, DTMB Report 1378, October 1959

6. Clement, E.P. and J.D. Pope, Graphs for Predicting the Resistance of Large Stepless Planing Hulls at High Speeds, DTMB Report 1318, April 1959

7. Clement, E.P. and J.D. Pope, Stepless and Stepped Planing Hulls, Graphs for Performance Prediction and Design, DTMB Report 1490, January 1961

8. Grumman Aircraft Engineering Corp., Study of Hydrofoil Seacraft, October 1958

FIG. B-I PLANING HULLS
APPROXIMATE HULL DIMENSIONS AS A FUNCTION OF TOTAL DISPLACEMENT

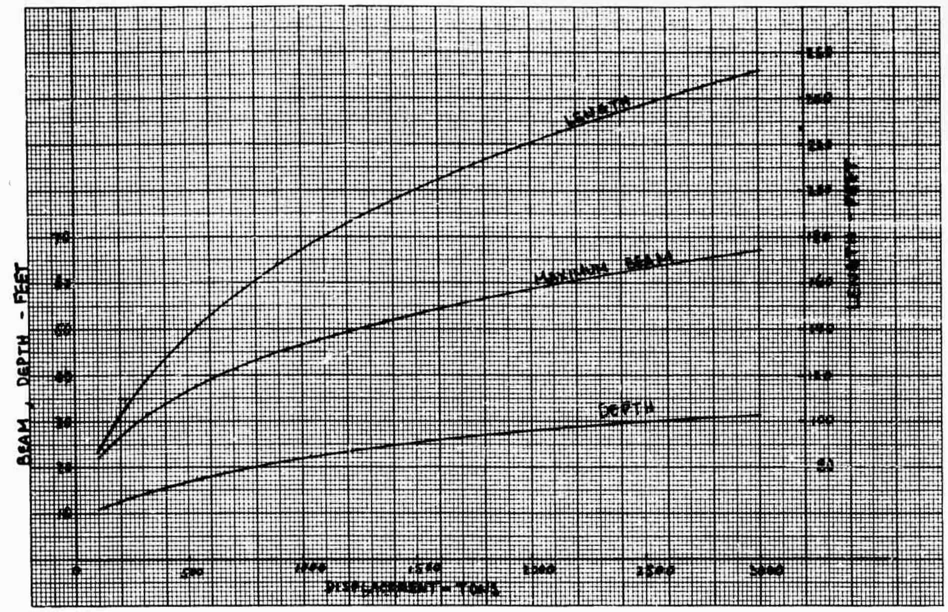


FIG. B-2 PLANING HULLS
SPECIFIC RESISTANCE VERSUS VOLUMETRIC DISPLACEMENT

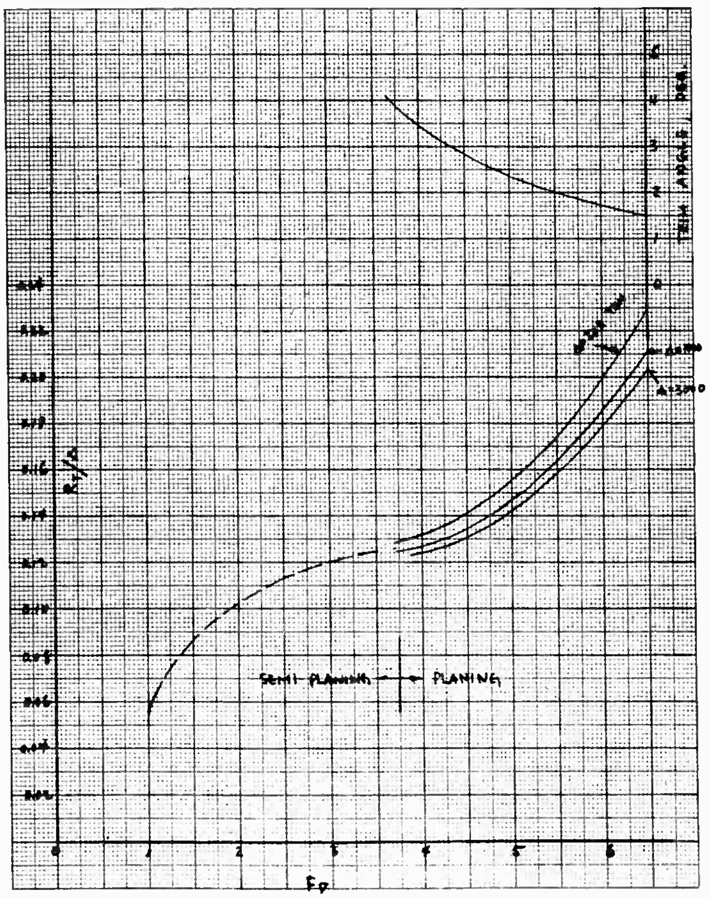


FIG. B-3 PLANING HULLS
SHAFT HORSEPOWER VERSUS SPEED FOR A RANGE OF DISPLACEMENT

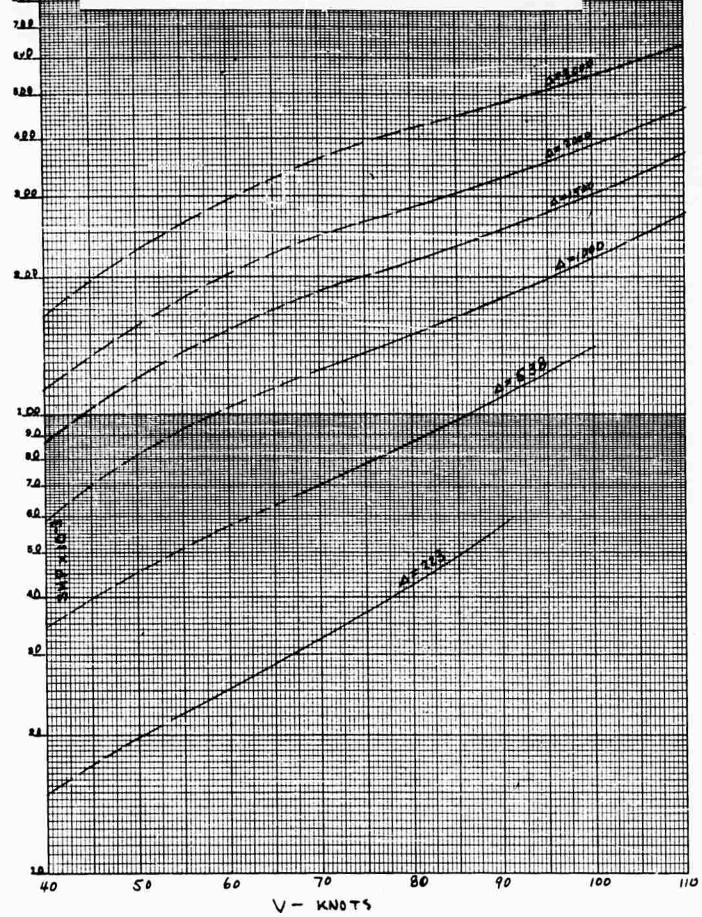


FIG. B-4 PLANING HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS
AS PERCENTAGE OF FULL LOAD DISPLACEMENT, NUCLEAR POWER PLANT
(Speed, 40 to 80 Knots)

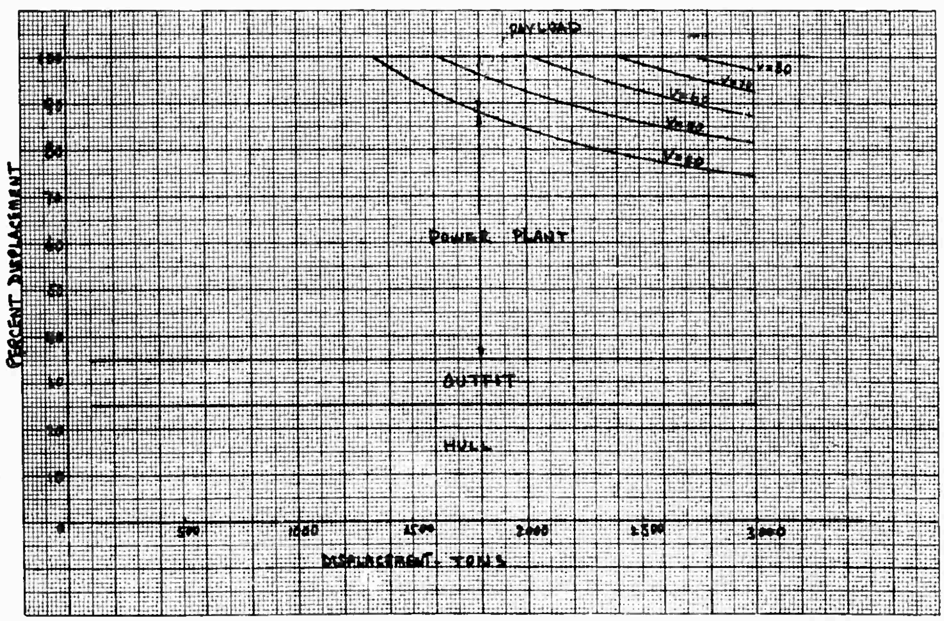


FIG. B-5 PLANING HULLS
DISPLACEMENT AND PAYLOAD VERSUS SPEED AND
REQUIRED SHAFT HORSEPOWER, GAS TURBINE POWER PLANT
(Range, 500 Nautical Miles)

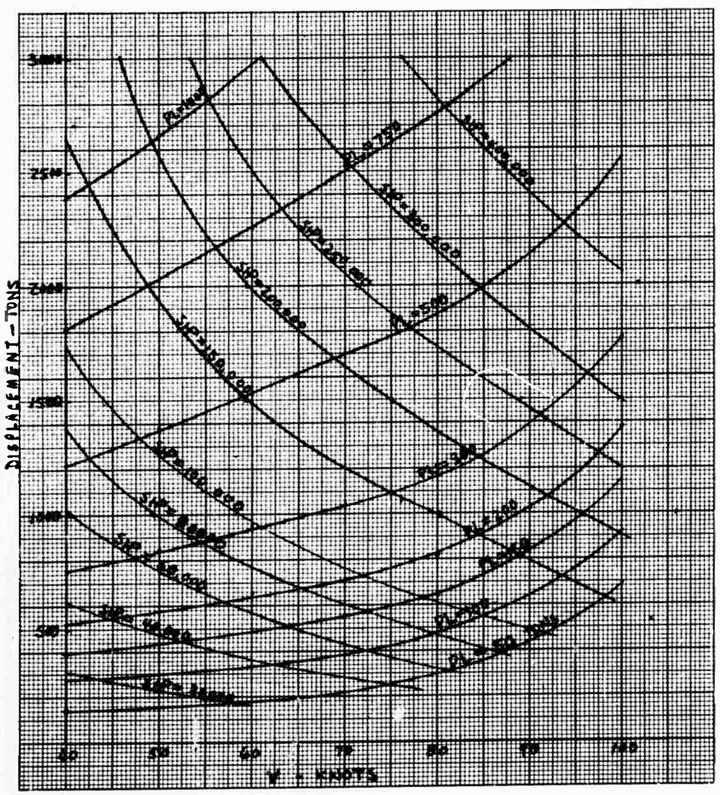


FIG. B-6 PLANING HULLS, DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT HORSEPOWER, GAS TURBINE POWER PLANT
(Range, 1,000 Nautical Miles)

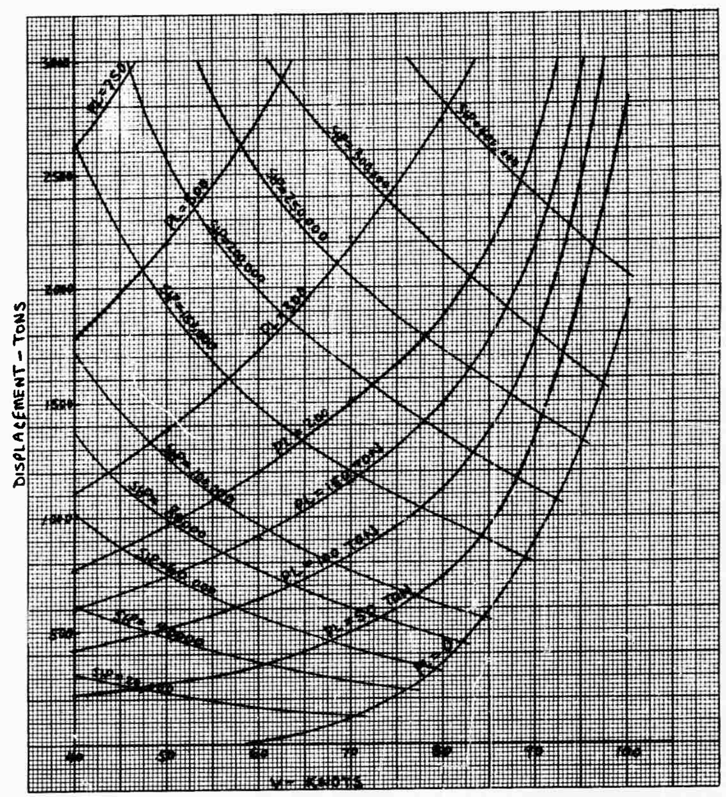


FIG. B-7 PLANING HULLS, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER, GAS TURBINE POWER PLANT
(Range, 1,500 Nautical Miles)

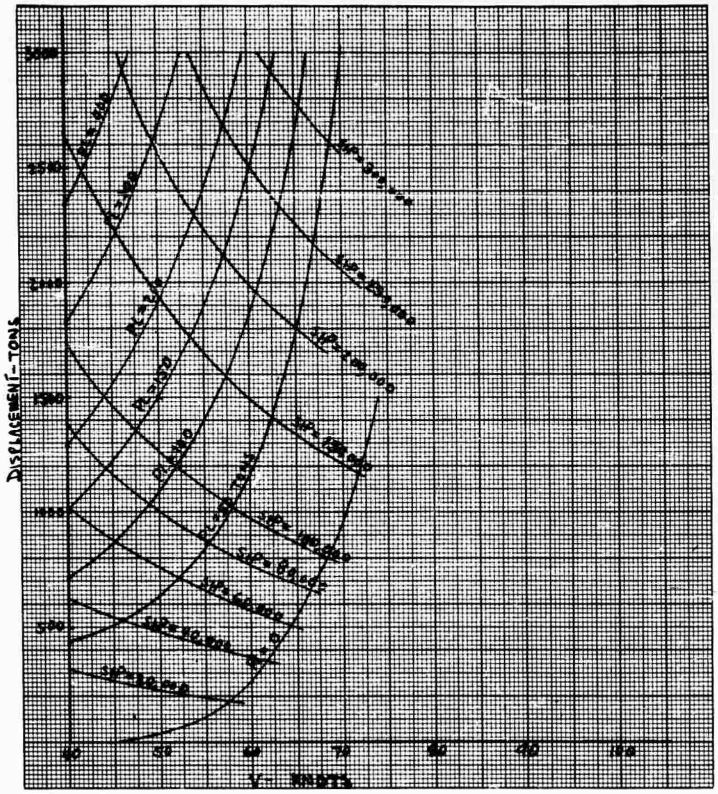


FIG. B-8 PLANING HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Speed, 40 Knots; Various Ranges)

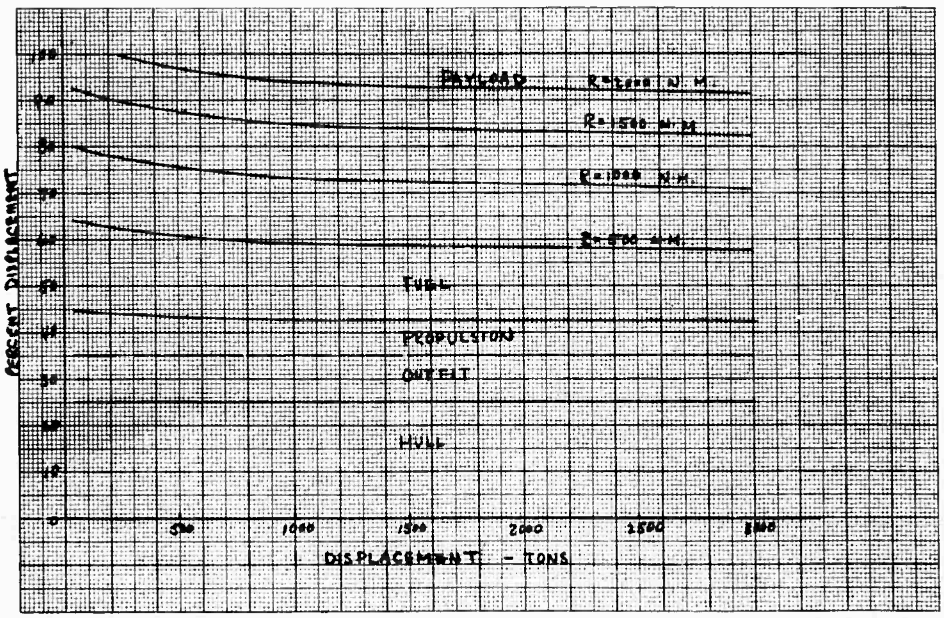


FIG. B-9 PLANING HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Speed, 50 Knots; Various Ranges)

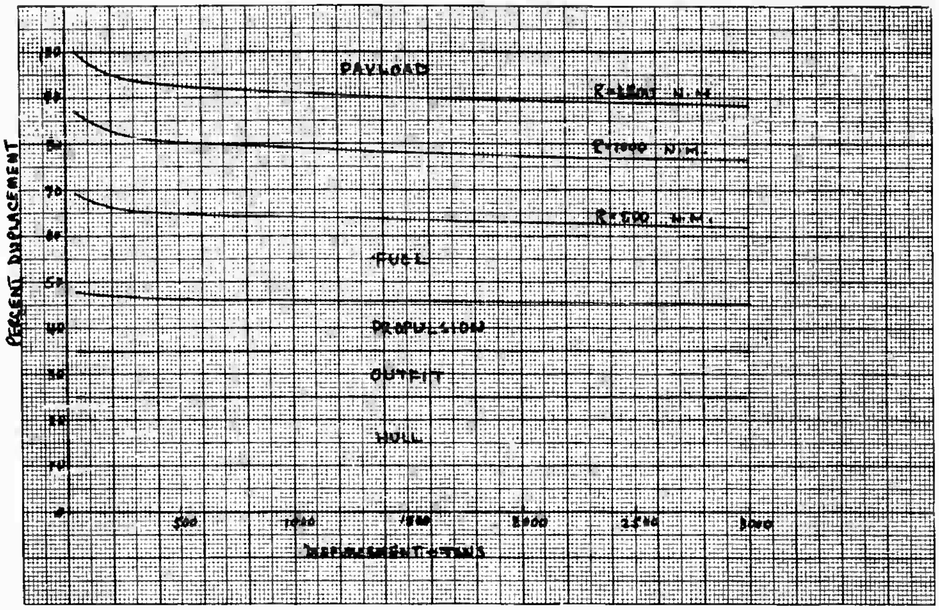


FIG. B-10 PLANING HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Speed, 60 Knots; Various Ranges)

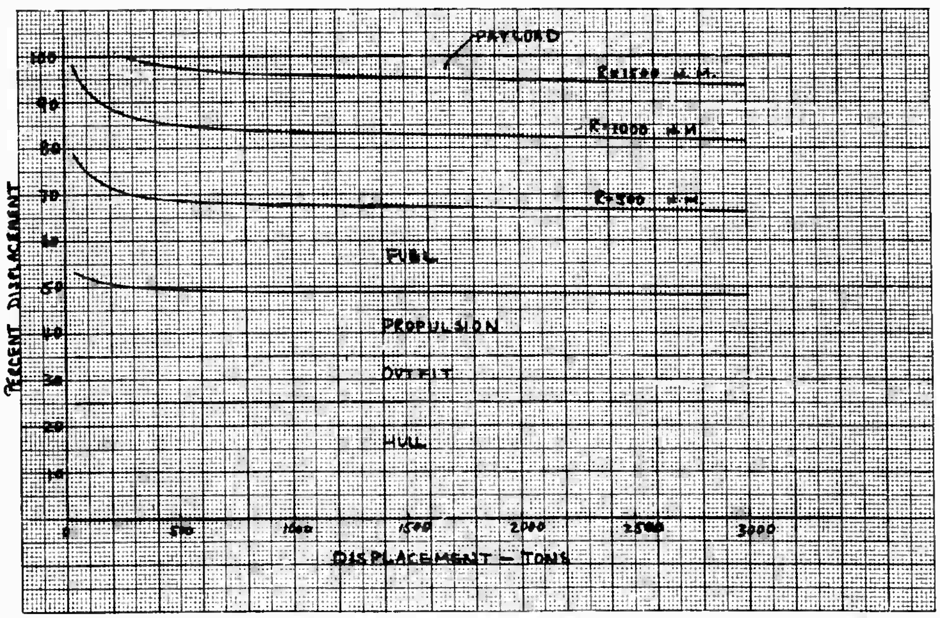


FIG. B-11 PLANING HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS
AB PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Speed, 70 Knots; Various Ranges)

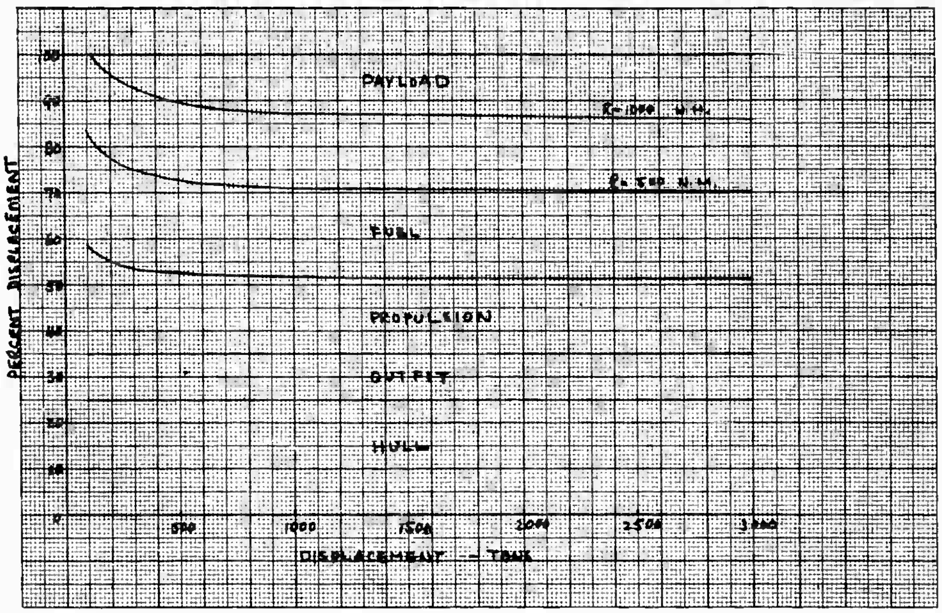


FIG. B-12 PLANING HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS
AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Speed, 80 Knots; Various Ranges)

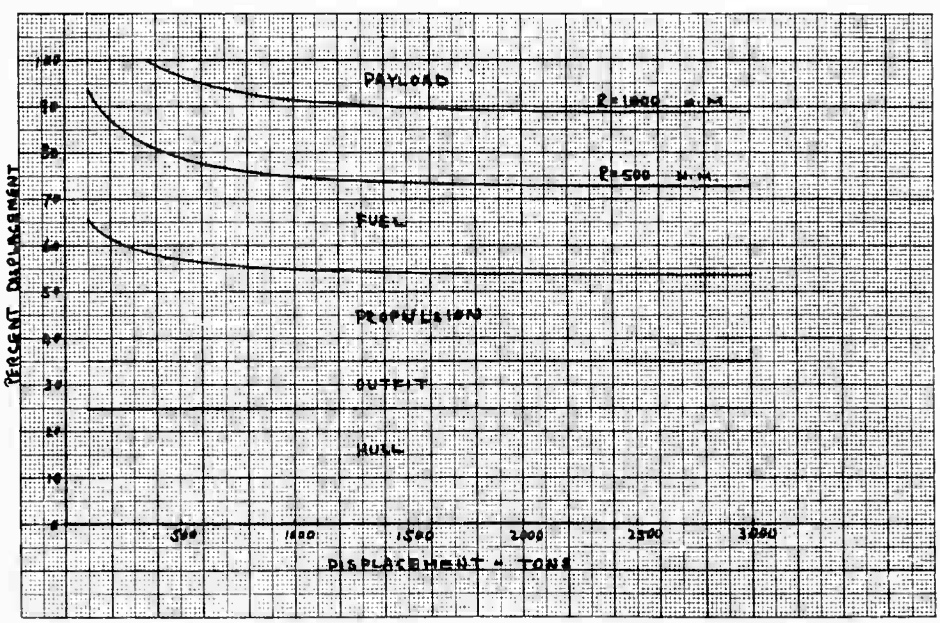


FIG. B-13 PLANING HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Speed, 90 Knots; Various Ranges)

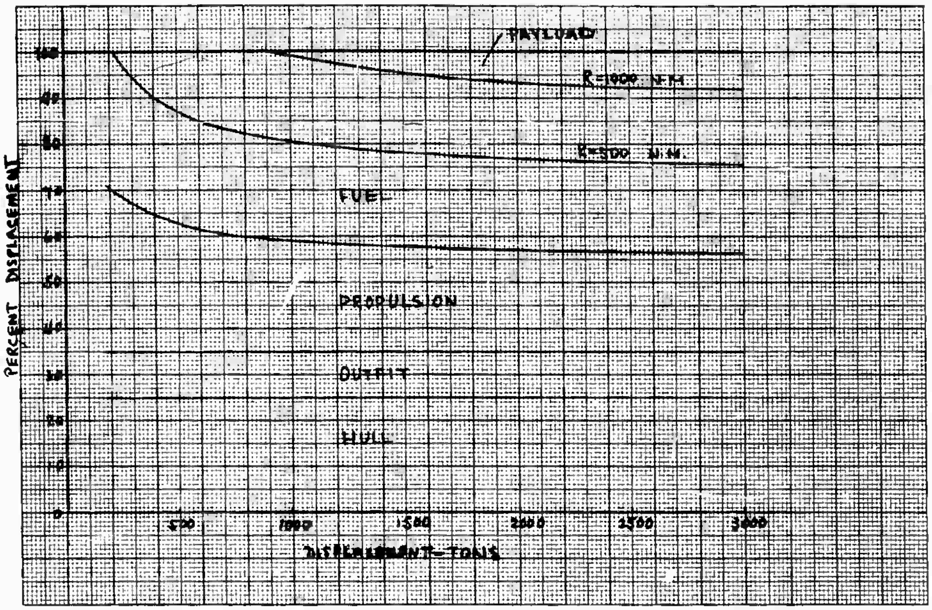
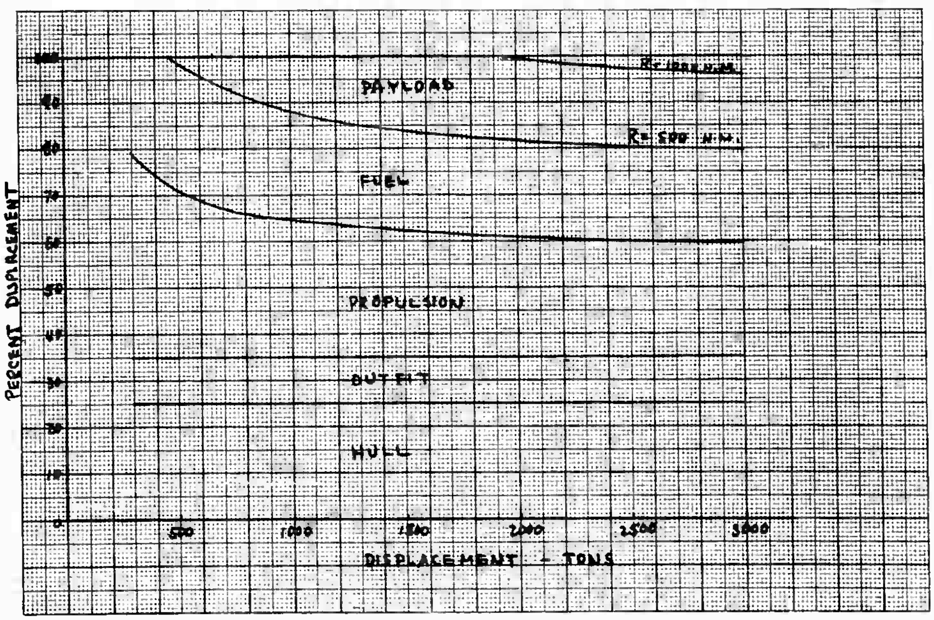


FIG. B-14 PLANING HULLS, PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Speed, 100 Knots; Various Ranges)



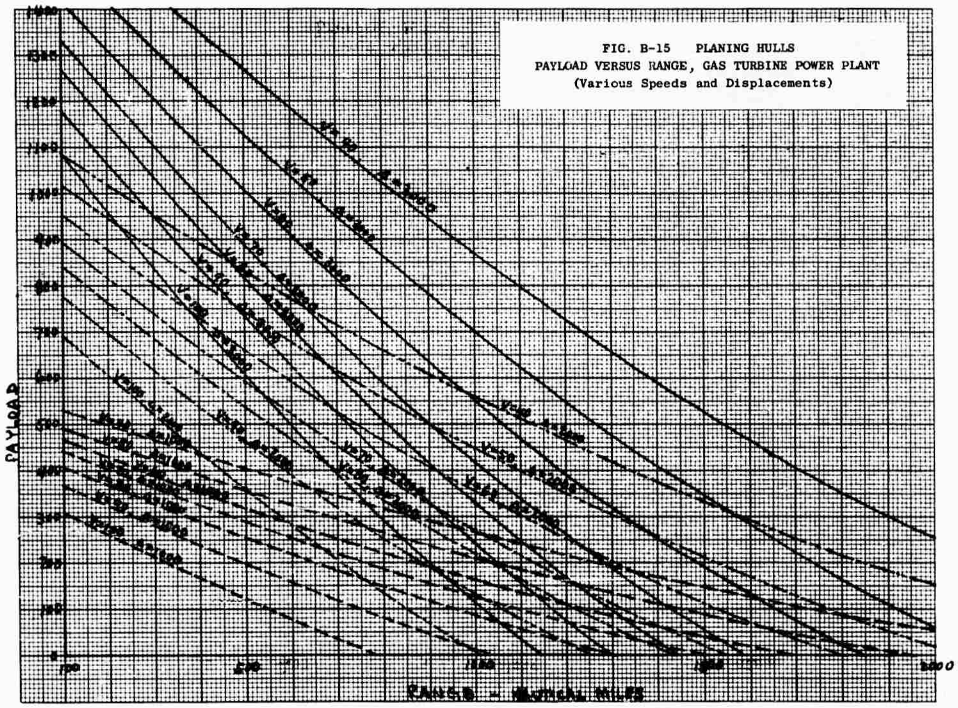


FIG. B-15 PLANING HULLS
PAYLOAD VERSUS RANGE, GAS TURBINE POWER PLANT
(Various Speeds and Displacements)

Appendix C

CONTENTS

TABLES

ILLUSTRATIONS

Illustrations (continued)

Illustrations (concluded)

Appendix C

HYDROFOILS

Appendix C

HYDROFOILS

Introduction

The purpose of this study is to provide information for assessing the feasibility of hydrofoils for amphibious fleet units in the 1970-1980 time period. A number of feasibility studies have predicted a promising future for seagoing hydrofoil crafts. However, a close look at these studies shows that inconsistent evaluations have been made in regard to technical details crucial to the prediction of the performance and, hence, to the potential uses of these crafts. This study does not attempt to develop original information but rather to assess, compare, and evaluate the available information and then, using best engineering judgment, to extract reasonable values with regard to the 1970-1980 time period.*

Vehicle Lift-to-Drag Ratio, L/D

Several studies have been made in an effort to predict the over-all vehicle lift-to-drag ratio as a function of cruise velocity and size of craft. Figure C-1 shows the results of a number of studies (designated simply as Study 1, Study 2, etc.), together with lines $L/D = 480/V$ and $L/D = 528/V$. Oakley[1] has suggested that the line $L/D = 480/V$ represents values for good practical designs of crafts of gross weight under 100 tons and cruise velocities of 40 to 80 knots. This line, obtained from statistical studies of existing crafts, also represents the state-of-the-art. It is clear that certain of the studies give rather optimistic predictions of achievable lift-to-drag ratios. It should be noted that L/D generally varies with cruise velocity and size, and there is discontinuity in the transitional area, that is, $V = 50 \sim 70$, between subcavitating and supercavitating crafts. The line $L/D = 528/V$ is a 10 percent improvement in L/D over the line suggested by Oakley, and has been selected to represent the L/D in this study for subcavitating crafts up to $V = 60$ knots and supercavitating crafts of $V = 60 \sim 100$ knots. This 10 percent increase is, of course, quite arbitrary; however, by comparison with the results of available studies, the line is believed to be a reasonable prediction for crafts in the 1970-1980 time period. No effort has been made to differentiate L/D for different sizes of craft. For large crafts, the line may appear to be somewhat conservative, since the L/D improves with the size of crafts.

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* Use has been made of the design studies of particular companies; however, specific reference to the sources of particular data has been purposely avoided.

Shaft Horsepower Required for Foilborne Operation

The shaft horsepower (SHP) required can be expressed as:

$$SHP = \frac{D\ V\ (1.689)}{550\ (PC)} \tag{1}$$

where

D = total drag in pounds

V = cruise velocity in knots

PC = total propulsive coefficient.

From L/D = 528/V (see Fig. C-1), the drag will be

$$D = LV/528 \quad . \tag{2}$$

Hence, by combining Eqs. (1) and (2),

$$SHP = \frac{LV^2\ (1.689)}{(550)(528)(PC)} \quad . \tag{3}$$

But $\Delta = L/2240$ where Δ = gross weight of craft in tons.

Hence,

$$SHP = \frac{\Delta V^2\ (1.689)(2240)}{(550)(528)(PC)} \tag{4}$$

or,

$$\frac{SHP}{\Delta} = \frac{0.013\ V^2}{(PC)} \quad . \tag{5}$$

One of the studies utilized shows that the over-all propulsive coefficient for supercavitating water propellers is 0.60 at V = 40 knots to 0.71 at V = 100 knots (see Fig. 7 of Reference No. 2). Using the figures for PC suggested in this study, the specific shaft horsepower, SHP/Δ, for a number of cruise velocities is calculated and plotted as a function of the velocity, as shown in Fig. C-2. Figure C-3 shows the comparison of the line of Fig. C-2 with other studies and points of existing crafts.

The shaft horsepower obtained above is the continuous rating horsepower required for the foilborne cruise condition. At take-off, the hydrofoil requires about 25 percent to 30 percent more power in calm water and approximately 50 percent more in rough sea conditions. The time required for take-off is less than 60 seconds. Hence, the prime

mover selected should be able to develop a maximum SHP of 1.5 times the continuous SHP for a duration of 60 seconds. Use of aircraft-type gas turbines has been assumed in developing the basic power plant weight estimates. It is also noted that lightweight nuclear power plants may prove to be promising for larger craft.[3] However, in the time span under consideration, it is believed that the gas turbine is the more feasible choice, considering engine efficiency, weight, and specific fuel consumption. Figure C-4 shows the specific weight of the foilborne propulsive plant (including prime mover, transmission, and propeller) per SHP plotted as a function of the cruise velocity.

Power Plant for Hullborne Operation

A lightweight diesel engine is chosen as the power plant for hullborne operations. Figure C-5, based on information in one study, shows the specific horsepower (SHP/$\triangle$) and specific power plant weight ($W_{HP}/\triangle$) as a function of the craft gross weight at the hullborne cruise velocity of 12 knots in calm water. It is noted that, for small crafts, the hullborne propulsive plant weight is a considerable fraction of the total propulsive plant (foilborne and hullborne) weight (W_{PR}) but, for large crafts (say for crafts of greater than 500 tons gross weight), this weight fraction is quite small (see Fig. C-6.)

Weight Estimates

Propulsive Plant Weight, W_{PR}

The propulsive plant includes prime movers, transmissions, and propellers for both foilborne and hullborne operations. Figure C-6 shows the specific propulsive plant weight, $W_{PR}/\triangle$, as a function of the cruise velocity. This figure combines information taken from Figs. C-2 through C-5.

Hull Weight, W_H

Figure C-7 shows a comparison of the hull weight estimates given in the various studies. The material is aluminum alloy. For the hull weight estimates of large crafts, Study No. 1 has adapted the structural weights of fast displacement vessels, such as destroyers (assuming that the structure of these vessels was replaced by a structure of aluminum alloy of equivalent strength), whose weight was found to be approximately 40 percent of the equivalent steel structure. It is believed that the figure of 40 percent is on the optimistic side. Hull weight was estimated in accordance with Study No. 2 shown on Fig. 7.

Weight of Foils and Struts, W_{FS}, for Fully Submerged Foils

Grumman[2] has made an extensive study of the design of foils and struts. In regard to the design criteria for strut length and strength, Grumman has indicated that the 20-foot wave was chosen as the sea state in which craft of this study would be required to operate while foil-borne without hull wave impact. The foil and strut structure is designed to carry simultaneous ultimate loads of 2.4 g's vertically and 0.75 g's in the lateral direction.

The results of this study are shown in Fig. C-8. The material of the foils and struts is titanium with an ultimate strength of 180,000 psi.

Outfit Weight, W_{AX}

This category includes: (1) basic electric group, (2) auxiliary equipment, including equipment for heating, ventilation, plumbing, fire fighting, steering, mooring winches, and so forth, and (3) outfit and furnishings, including hull fittings, nonstructural bulkheads, painting, deck covering, lockers, galley equipment, living space furnishings, and so forth.

Figure C-9 shows some studies on the specific weight of this category, W_{AX}/Δ, plotted as a function of displacement. The line from Study No. 3 has been selected for use in deriving weight estimates in this appendix.

Fuel Weight, W_F

$$R = \frac{325\ (L/D)_{total}\ (PC)}{SFC} \log_e \left[\frac{1}{1 - \frac{W_F}{\Delta}} \right] \quad \text{(Breguet's Range Equation)}$$

where

R = range in nautical miles and

SFC = specific fuel consumption in lbs/SHP/hr.

Transposing this equation, one gets

$$\frac{W_F}{\Delta} = 1 - \frac{1}{e^x} \qquad (6)$$

where

$$x = \frac{R\ (SFC)}{325\ (L/D)_{total}\ (PC)} \quad .$$

For aircraft gas turbines, PC can be assumed as 0.65 throughout for simplicity; SFC is given as 0.55 lbs/SHP/hr; and $(L/D)_{total} = \frac{528}{V}$ as chosen before. Then,

$$x = \frac{R\ (0.55)\ V}{(0.65)(528)(325)} = 4.93 \times 10^{-6}\ R\ V \quad . \qquad (7)$$

Using Eqs. (6) and (7), the specific fuel weight, W_F/Δ, was calculated. The results, plotted as a function of cruise velocity, are shown in Fig. C-10.

Hull Size

Table C-I shows data for some recently built or optimized design study hydrofoils and destroyers. By comparing form factors given in Table C-I, it may be said that the length of the seagoing hydrofoil is approximately 20 percent greater than that of the destroyer of the same displacement, and the hull form is, in general, fuller than that of the destroyer. In order to give an idea of the hull size of seagoing hydrofoils, Fig. C-11 has been prepared by assuming $Vol/L^3 = 1.0 \times 10^{-3}$; $B/L = 0.14$; and $B/H = 2.5$.

Summary Curves: Non-Nuclear Propulsion

The information obtained so far is grouped and summarized in Figs. C-12 through C-23, from which an appraisal of potential seagoing hydrofoils in the 1970-1980 time period can be readily made.

The assumptions made are restated below:

1. Design sea state: wave height = 20' (equivalent to sea state of 5-6).

2. Lift-to-drag ratio: L/D = 528/V.

Table C-I

DESIGN STUDY DATA FOR SOME RECENTLY BUILT OR OPTIMIZED HYDROFOILS AND DESTROYERS

Item	Hydrofoils		Grumman* Optimum Design	Grumman** Optimum Design	Destroyers			
	PC (H)	AG (EH)			DDG 2	DD 945-951	DDE 445	DD 421
L (over-all)	115'-9"	220'-5"	251'-0"	322'-0"	431'-0"	418'-6"	376'-6"	348'-2"
B (molded)	31'-1"	40'-5"	37'-0"	46'-0"	47'-0"	45'-0"	39'-4"	35'-4"
H (draft), hullborne	6'-0"	6'-1"	10'-0"	29'-0"	20'-0"	20'-0"	18'-0"	18'-0"
Displacement (tons)	110	300	500	1,000	4,500	4,050	3,040	2,580
B/L	0.268	0.183	0.147	0.143	0.109	0.107	0.105	0.102
B/H	5.18	6.65	3.7	1.58	2.35	2.25	2.2	1.96
Vol/L^3	2.48×10^{-3}	0.98×10^{-3}	1.1×10^{-3}	1.05×10^{-3}	1.97×10^{-3}	1.94×10^{-3}	2×10^{-3}	2.14×10^{-3}

* Gas turbine.

** Nuclear power.

3. Propulsion plant includes:

Main power plant for foilborne operation

a. Aircraft-type gas turbine (modified for marine use)
b. Transmission
c. Water propellers - supercavitating

Auxiliary power plant for hullborne operation

a. Lightweight diesel engine
b. Transmission
c. Water propellers

4. Aluminum alloy hull structure

5. The weights included in the category of outfit are:

a. Basic electric group
b. Auxiliary systems, including equipment for heating, ventilation, plumbing, fire fighting, steering, mooring winches, etc.
c. Outfit and furnishings, including hull fittings, nonstructural bulkheads, painting, deck covering, lockers, galley equipment, living space furnishings, and so forth.

6. Foils and struts are made of titanium and are retractable.

7. Fuel weight has been calculated by:

$$\frac{W_F}{\Delta} = 1 - \frac{1}{e^x}$$

where:

$$x = \frac{R\ (SFC)}{325\ (L/D)_{total}\ (PC)}\ .$$

8. Payload = displacement - $(W_H + W_{PR} + W_{AX} + W_{FS} + W_F)$.

Summary Curves: Nuclear Propulsion

The nuclear power plant concepts discussed in one of the studies in point are the closed-cycle, gas-cooled reactor located (a) above water and (b) under water. To gain some insight into the feasibility of nuclear power for hydrofoils, the line for above-water compartment configuration used in that study was employed to construct Figs. C-24 and C-25. These figures, which show payload potential, illustrate the heavy relative weight of shielding required for nuclear propulsion. As to the relative merits of the two configurations (above water and under water), it was indicated that the specific weight of the pod is considerably less than the hull-mounted plant. However, performance with the under-water pod configuration would be reduced because of the water drag of the pod and its supporting strut. Therefore, the relative merits of the hull mounted and submerged pod configurations are not completely reflected in the comparison of their specific weights. Further investigation of the shielding and containment requirements might prove it possible to reduce shielding weight. We have assumed that the under-water pod configuration will not be feasible for the 1970 to 1980 time period.

To approximate nuclear hydrofoils greater than 3,000 tons, it has been assumed that the weight percentages shown on Figs. C-24 and C-25 for hull and auxiliaries would remain the same as at 3,000 tons displacement. Figs. C-26 through C-28 show payload potential of nuclear-powered hydrofoils up to 5,000 tons total displacement at 50 and 60 knots.

REFERENCES

1. Oakley, O.H., Hydrofoils - A State of the Art Summary, INA Proceeding of The National Meeting on Hydrofoil and Air Cushion Vehicles, September 1962

2. Geyer, L.A. and G.J. Wennagel, A Feasibility Study of Hydrofoil Seacraft, SNAME, Chesapeake Section

3. Grumman Aircraft Engineering Corp., Study of Hydrofoil Seacraft, Vols. I and II, October 1958

FIG. C-1 HYDROFOILS
LIFT/DRAG RATIO VERSUS CRUISE VELOCITY
(Water Propeller)

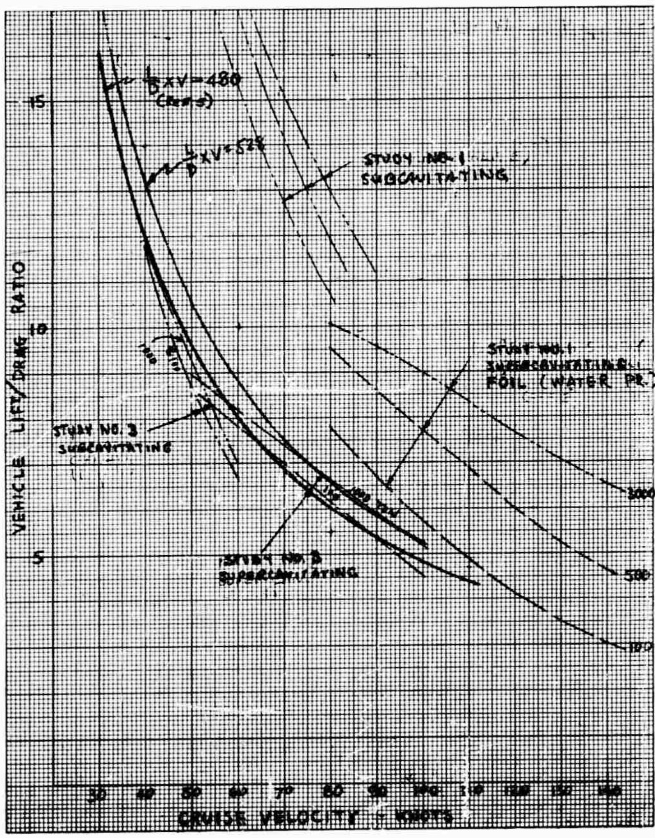


FIG. C-2 HYDROFOILS
HORSEPOWER FOR FOILBORNE OPERATION AT DESIGN SPEED

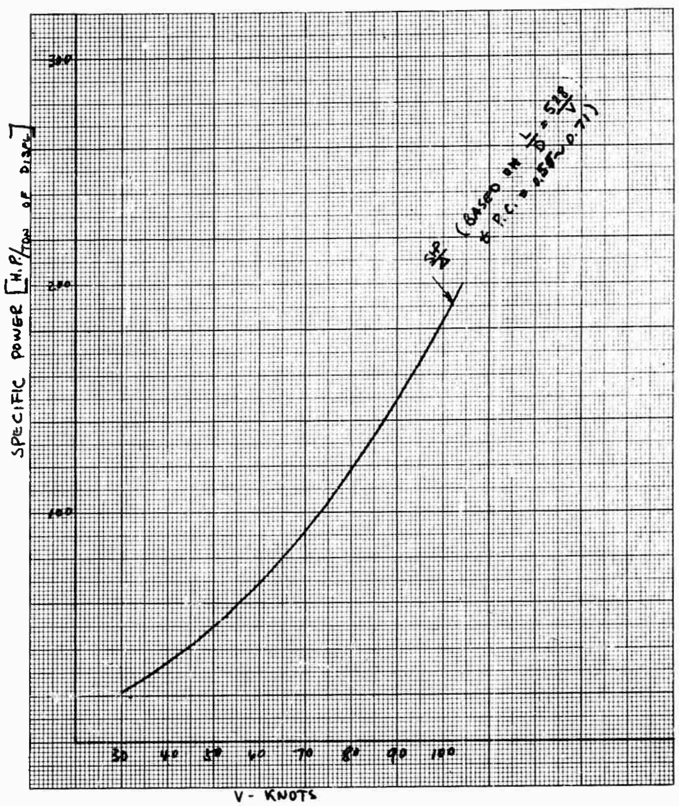


FIG. C-3 HYDROFOILS
COMPARISON OF DERIVED HORSEPOWER CURVE (FIG. C-2)
WITH HORSEPOWER CURVES DESIVED IN OTHER DESIGN STUDIES

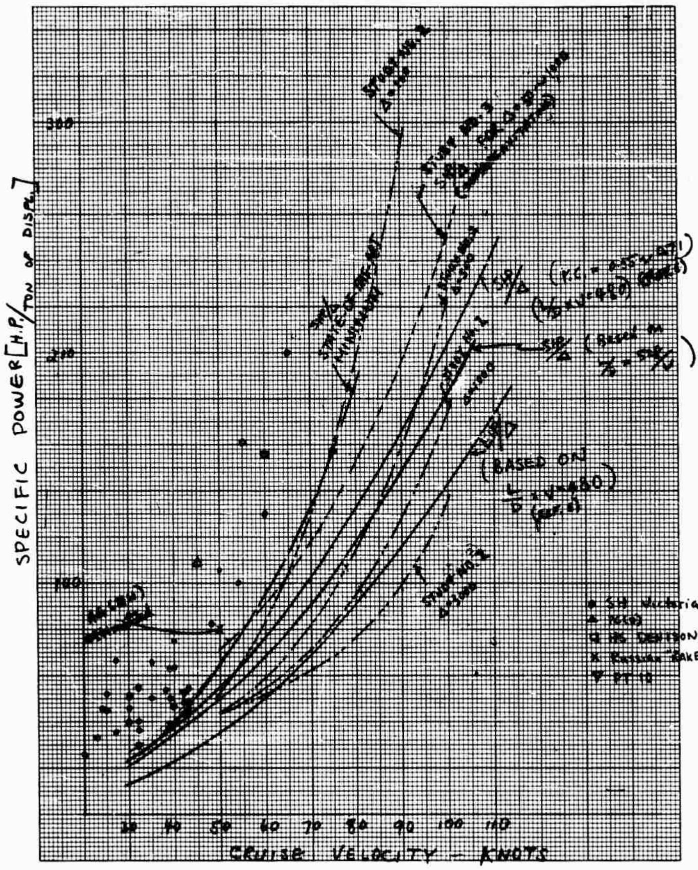


FIG. C-4 HYDROFOILS
WEIGHT OF PROPULSION PLANT FOR FOILBORNE OPERATION
VERSUS CRUISE VELOCITY

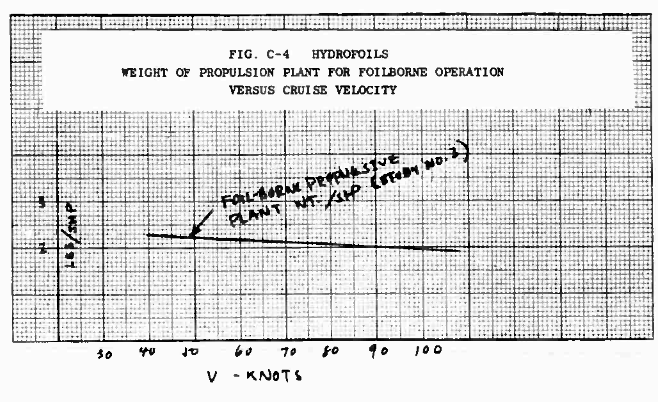


FIG. C-5 HYDROFOILS
SPECIFIC HORSEPOWER AND POWER PLANT WEIGHT
VERSUS DISPLACEMENT FOR HULLBORNE OPERATIONS

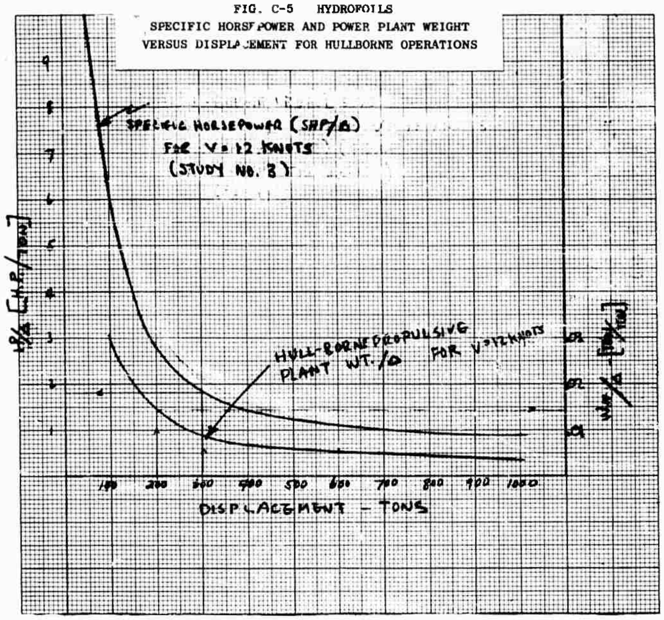
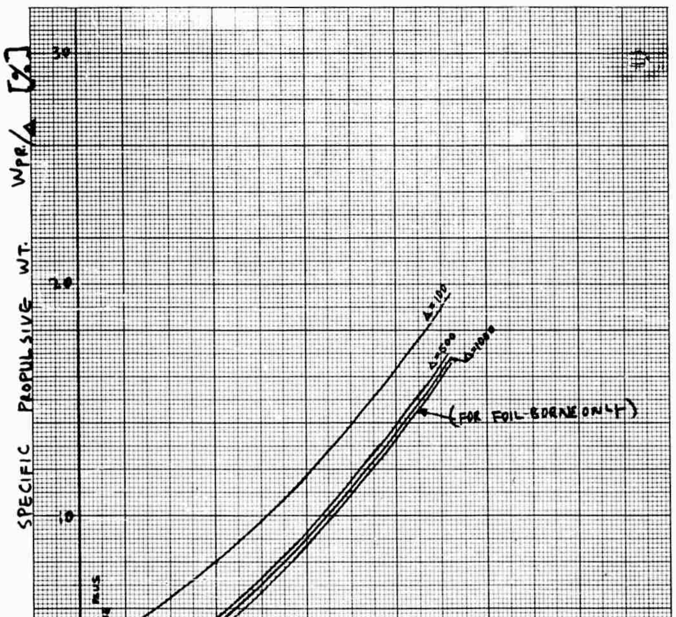


FIG. C-6 HYDROFOILS
TOTAL PROPULSION SYSTEM WEIGHT VERSUS SPEED
(L/D = 528/V; Various Displacements)



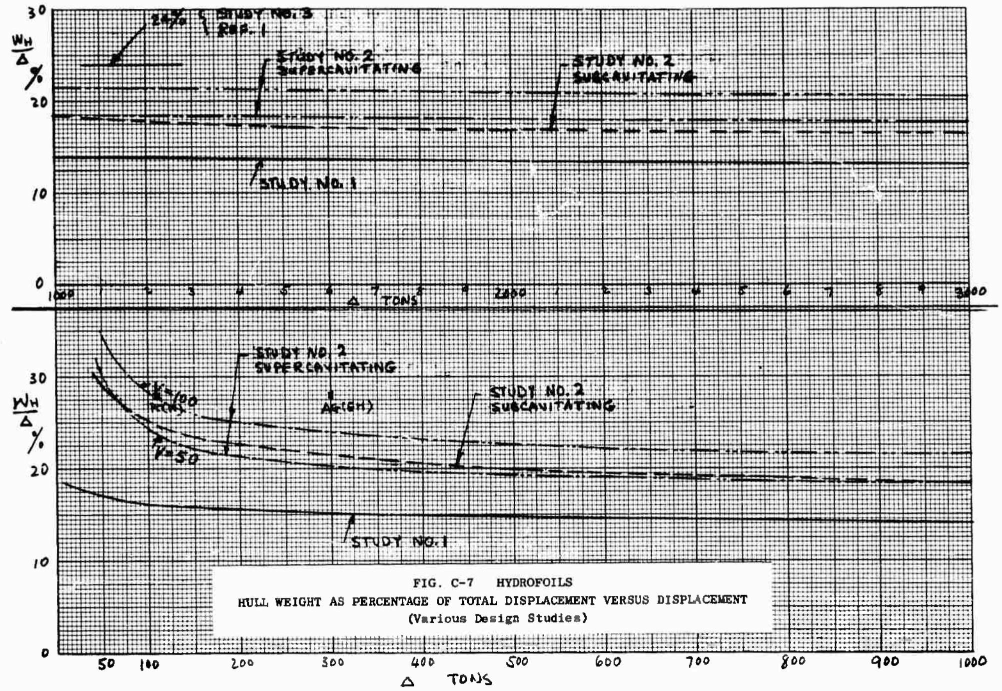


FIG. C-7 HYDROFOILS
HULL WEIGHT AS PERCENTAGE OF TOTAL DISPLACEMENT VERSUS DISPLACEMENT
(Various Design Studies)

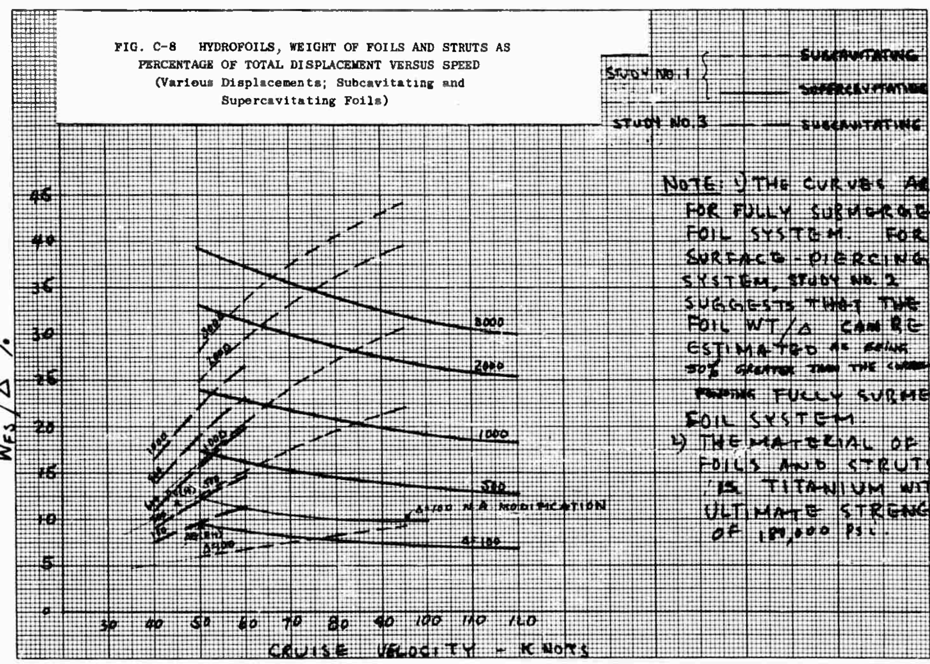


FIG. C-8 HYDROFOILS, WEIGHT OF FOILS AND STRUTS AS PERCENTAGE OF TOTAL DISPLACEMENT VERSUS SPEED
(Various Displacements; Subcavitating and Supercavitating Foils)

FIG. C-9 HYDROFOILS, OUTFIT WEIGHT AS PERCENTAGE OF TOTAL DISPLACEMENT
VERSUS TOTAL DISPLACEMENT

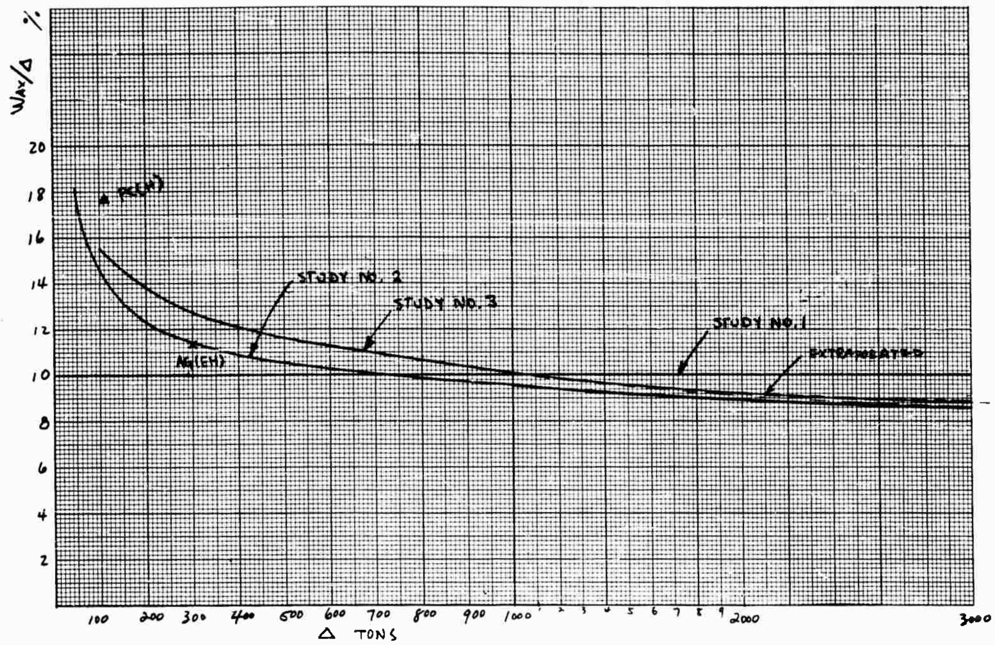


FIG. C-10 HYDROFOILS
FUEL WEIGHT AS PERCENTAGE OF TOTAL DISPLACEMENT VERSUS SPEED
(Various Ranges)

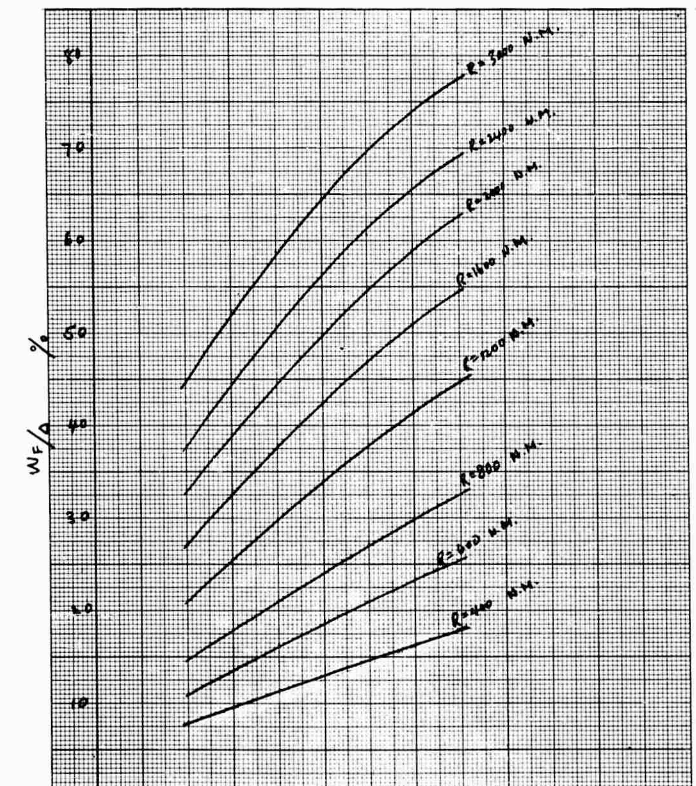


FIG. C-11 HYDROFOILS
LENGTH, BEAM, AND DRAFT AS A FUNCTION OF
TOTAL DISPLACEMENT

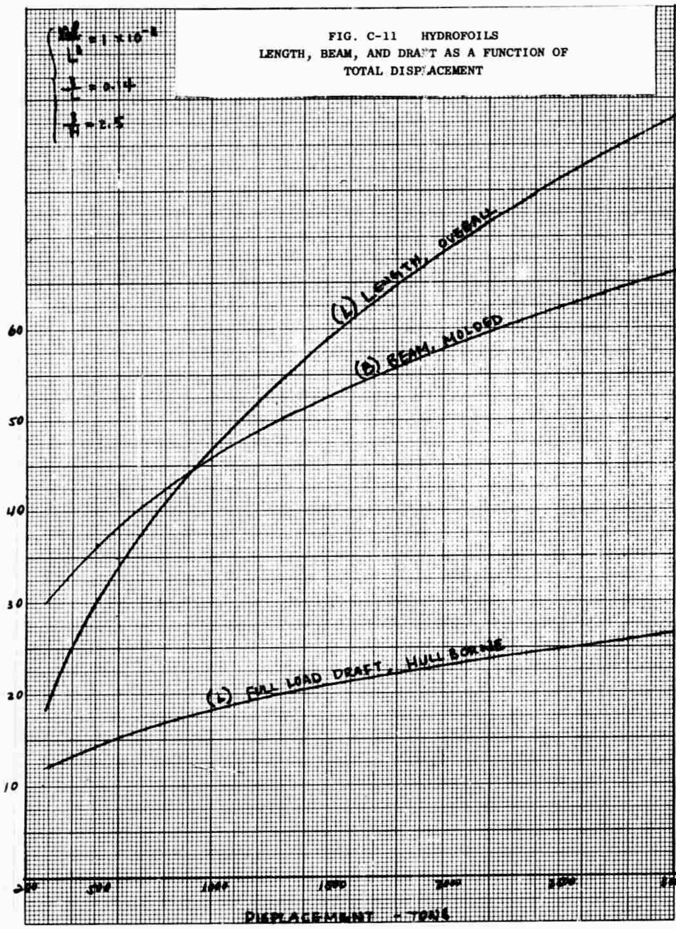


FIG. C-12 HYDROFOILS
DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED HORSEPOWER
(Range, 500 Nautical Miles)

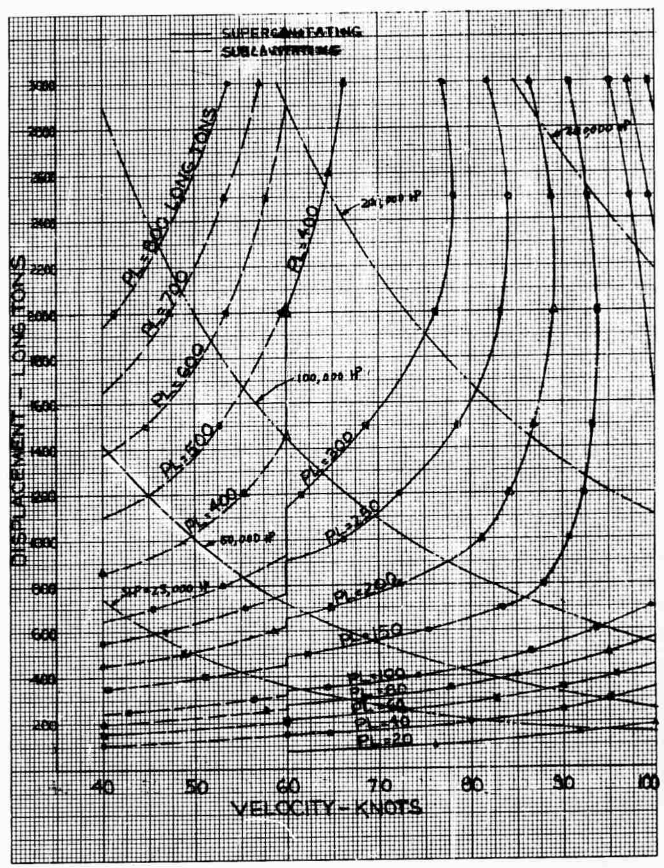


FIG. C-13 HYDROFOILS
DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED HORSEPOWER
(Range, 1,000 Nautical Miles)

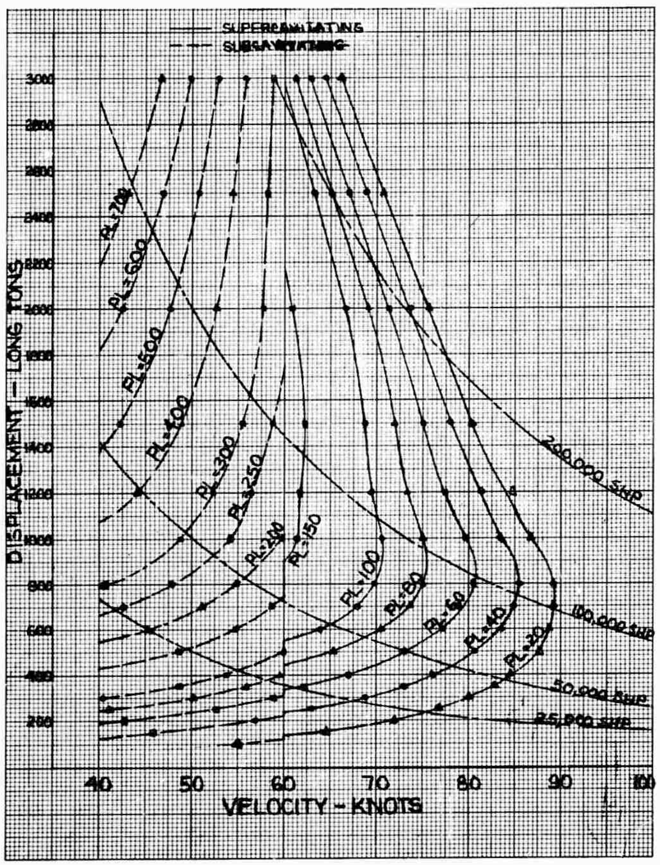


FIG. C-14 HYDROFOILS
DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED HORSEPOWER
(Range, 1,500 Nautical Miles)

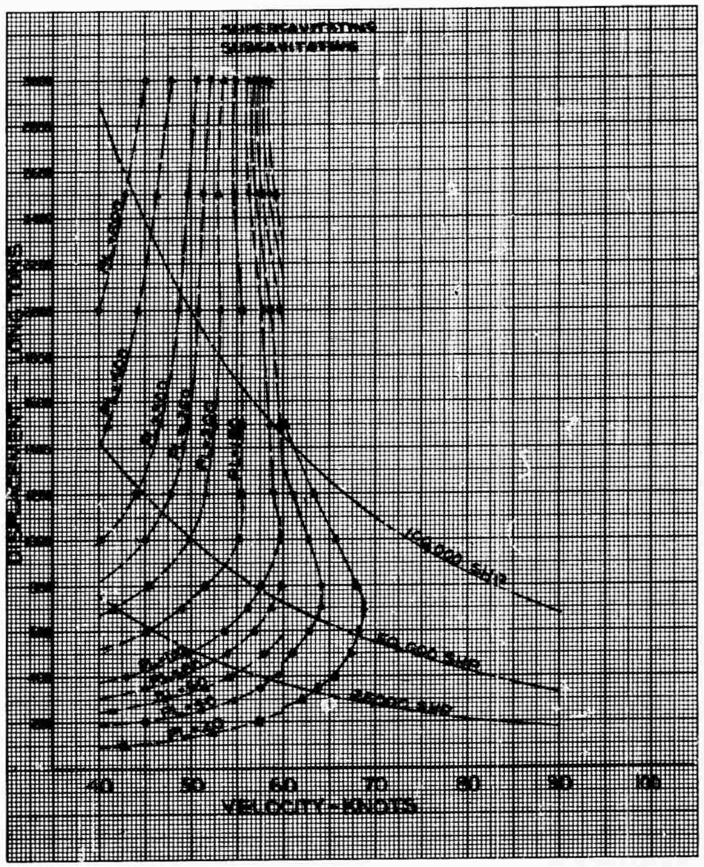


FIG. C-15 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 40 Knots; Subcavitating Foils)

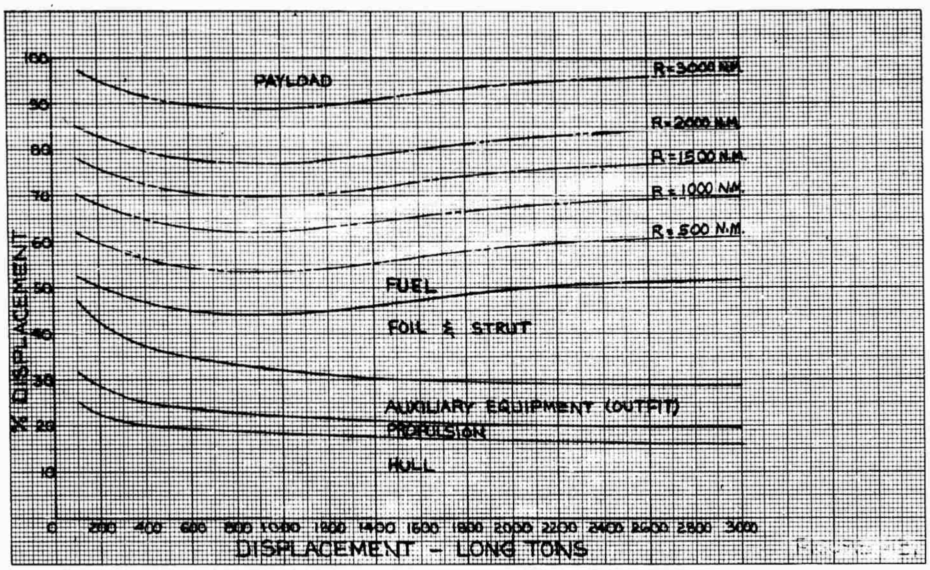


FIG. C-16 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 50 Knots; Subcavitating Foils)

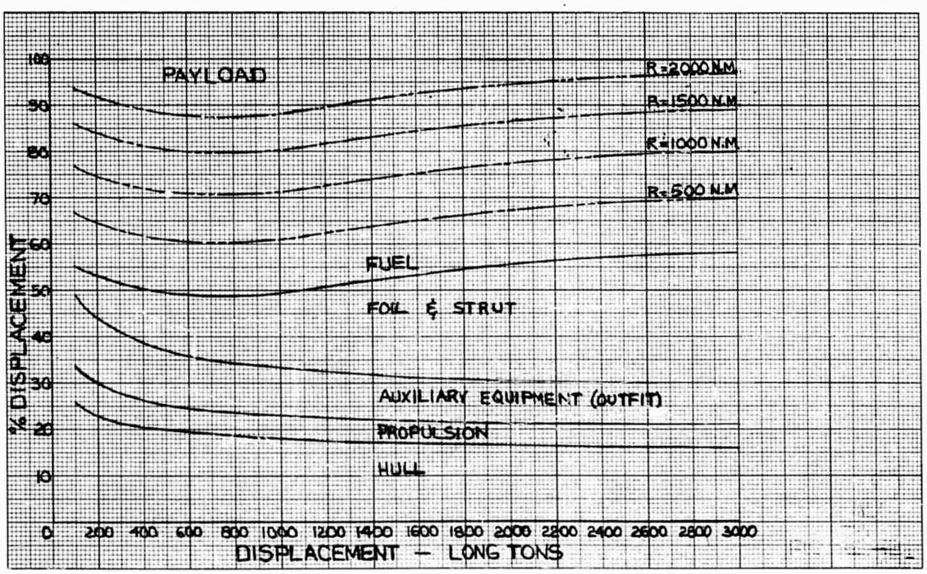


FIG. C-17 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 60 Knots; Subcavitating Foils)

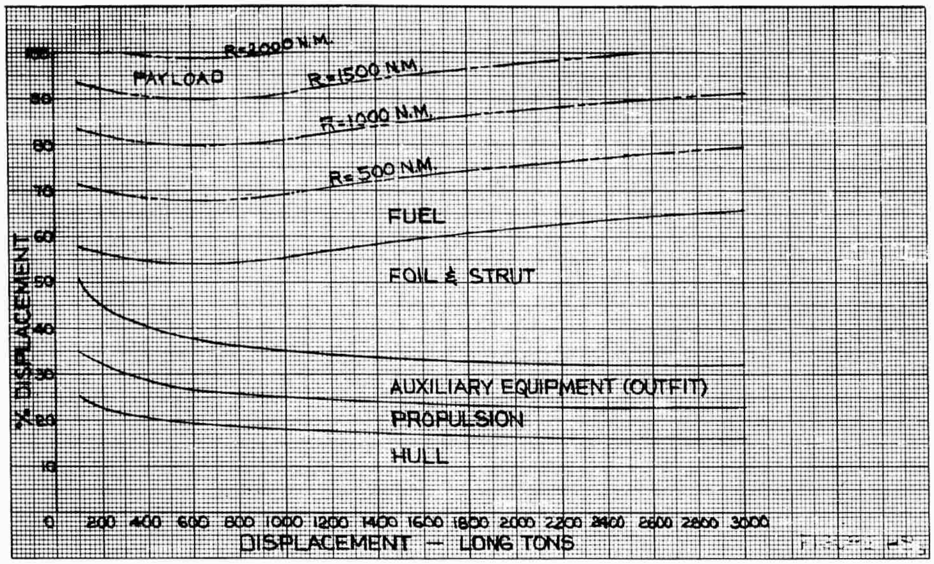


FIG. C-18 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 60 Knots; Supercavitating Foils)

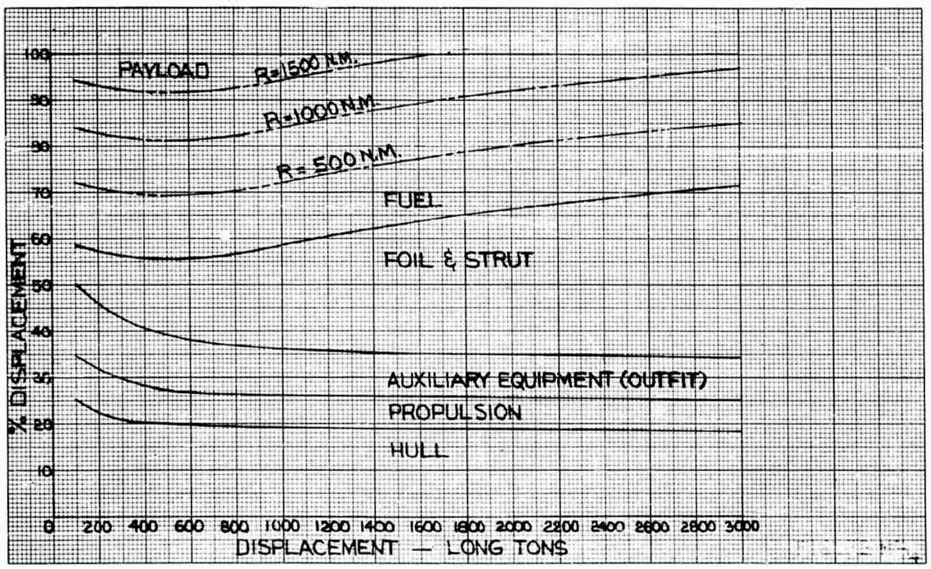


FIG. C-19 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 70 Knots; Supercavitating Foils)

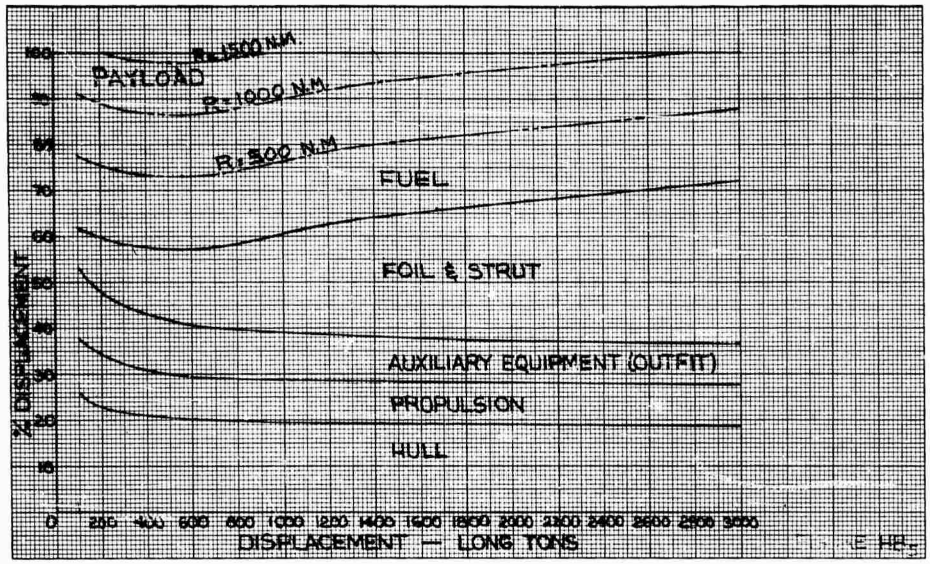


FIG. C-20 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 80 Knots; Supercavitating Foils)

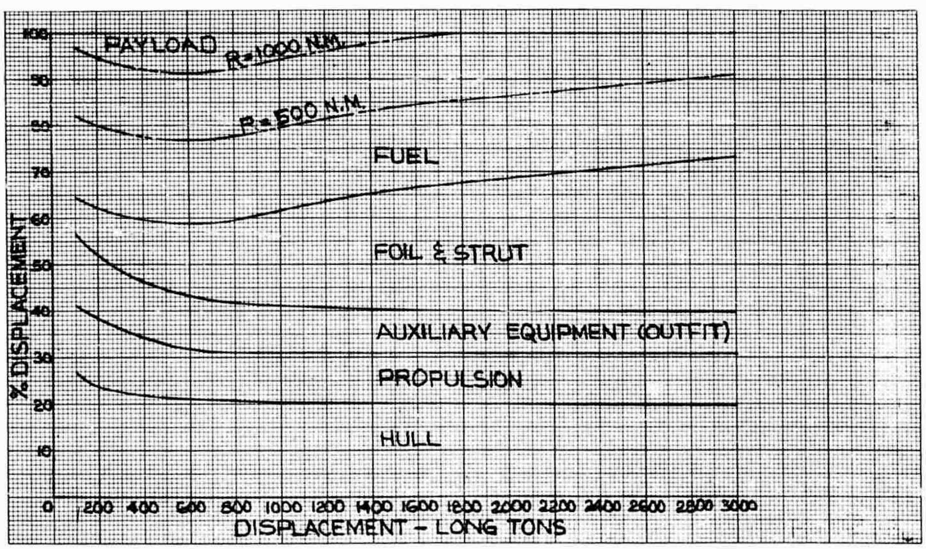


FIG. C-21 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 90 Knots; Supercavitating Foils)

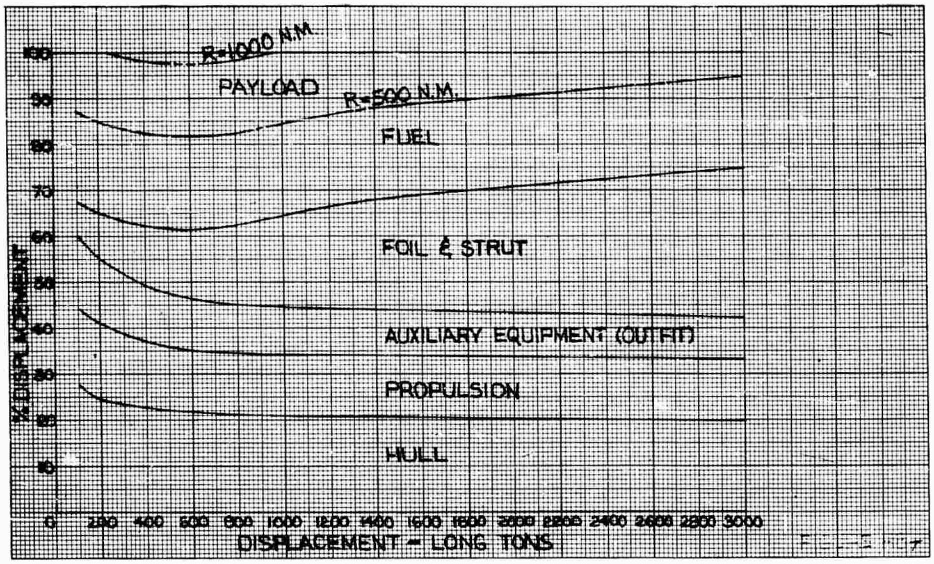


FIG. C-22 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT
(Speed, 100 Knots; Supercavitating Foils)

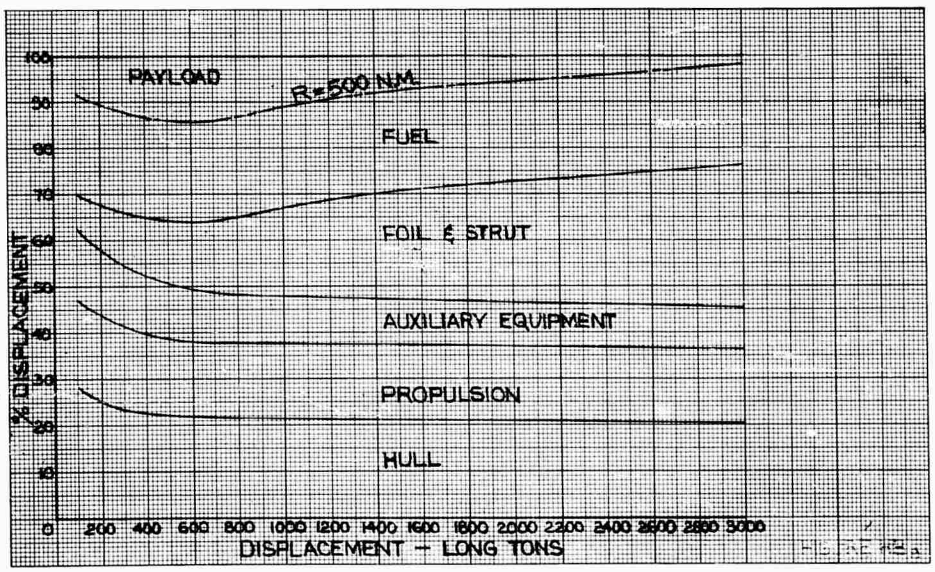


FIG. C-23 HYDROFOILS
PAYLOAD VERSUS RANGE, GAS TURBINE POWER PLANT
(Various Speeds and Displacements)

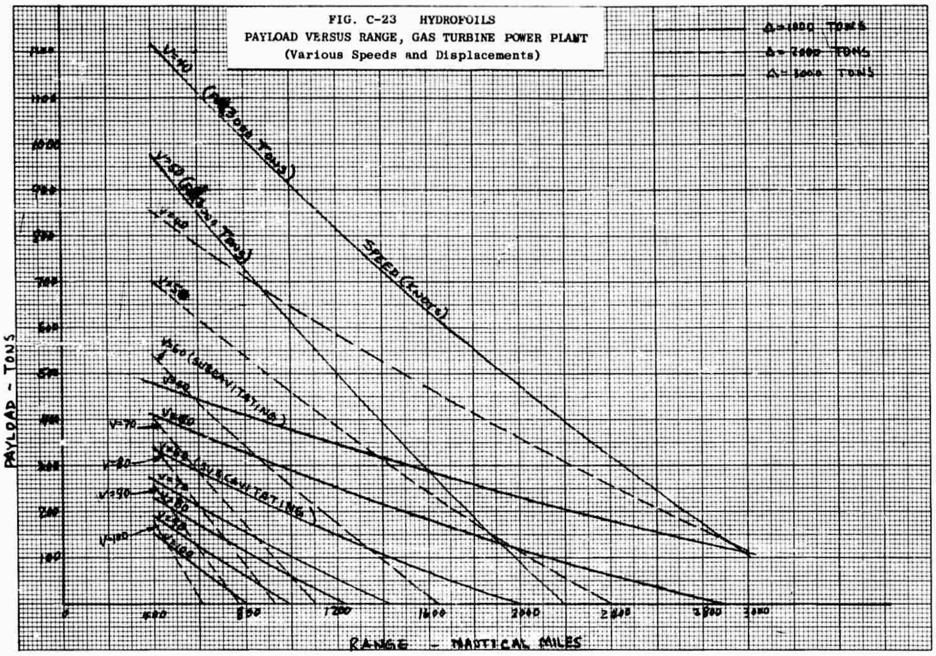


FIG. C-24 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE
OF FULL LOAD DISPLACEMENT, NUCLEAR POWER PLANT
(Speed, 40 Knots; Subcavitating Foils)

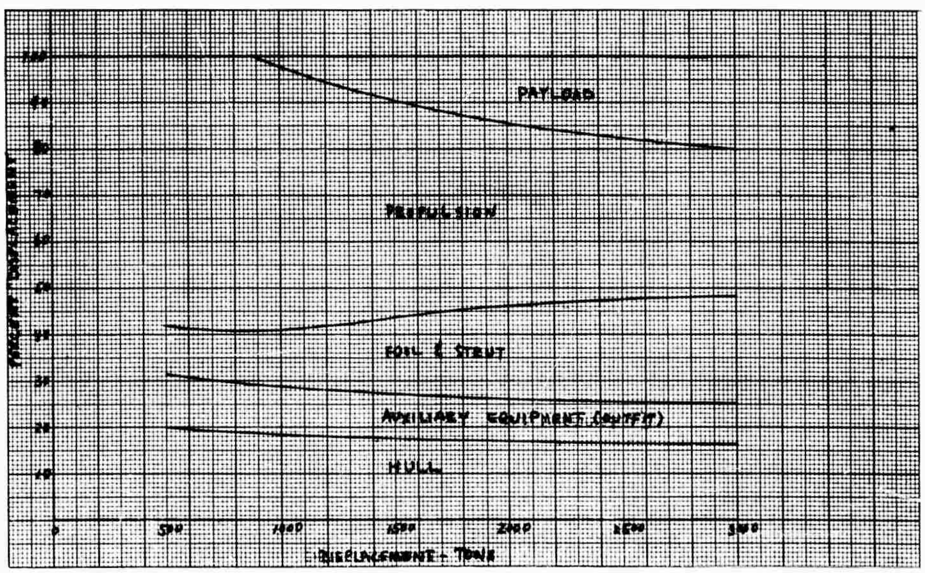


FIG. C-25 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE
OF FULL LOAD DISPLACEMENT, NUCLEAR POWER PLANT
(Speed, 50 Knots; Subcavitating Foils)

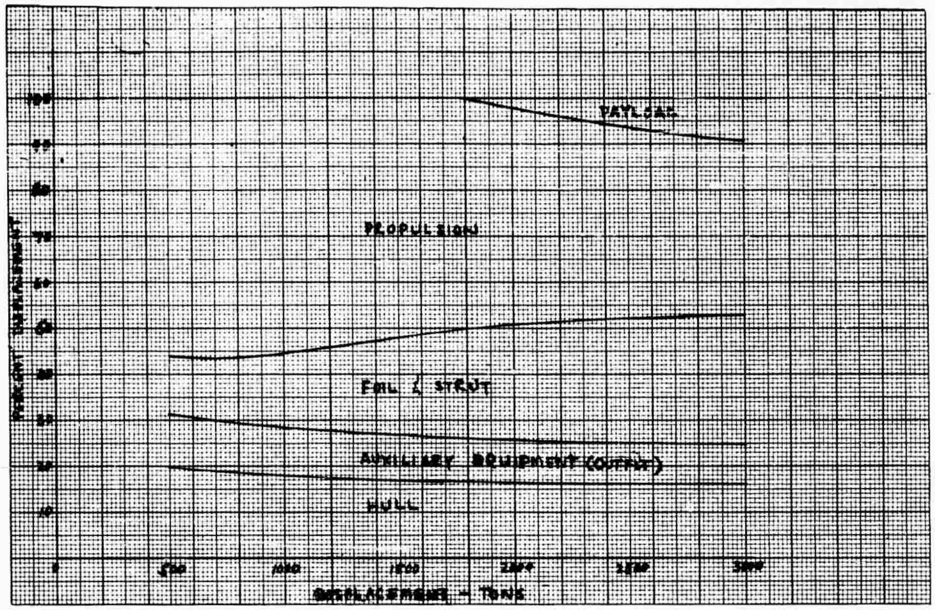


FIG. C-26 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE
OF FULL LOAD DISPLACEMENT, NUCLEAR POWER PLANT
(Speed, 60 Knots; Subcavitating Foils)

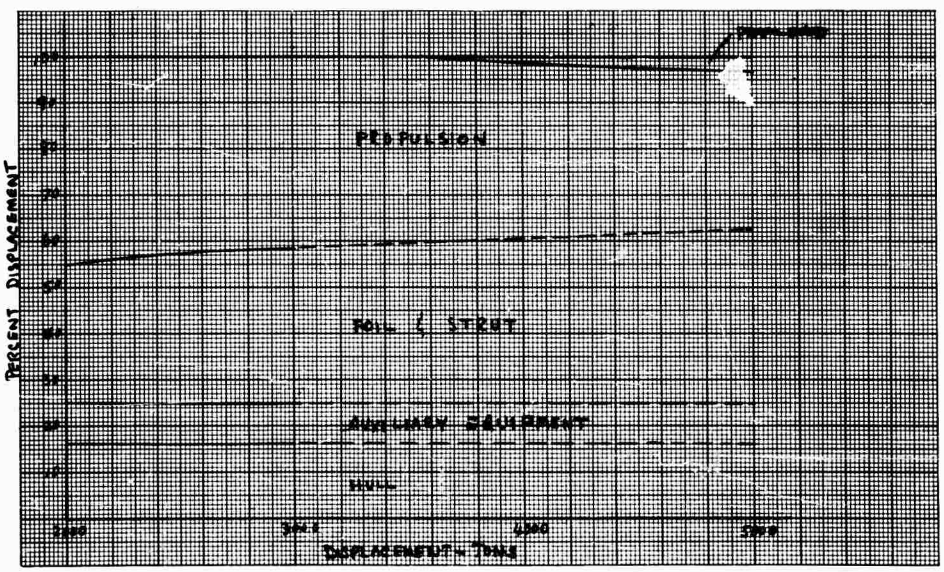


FIG. C-27 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE
OF FULL LOAD DISPLACEMENT, NUCLEAR POWER PLANT
(Speed, 50 Knots; Supercavitating Foils)

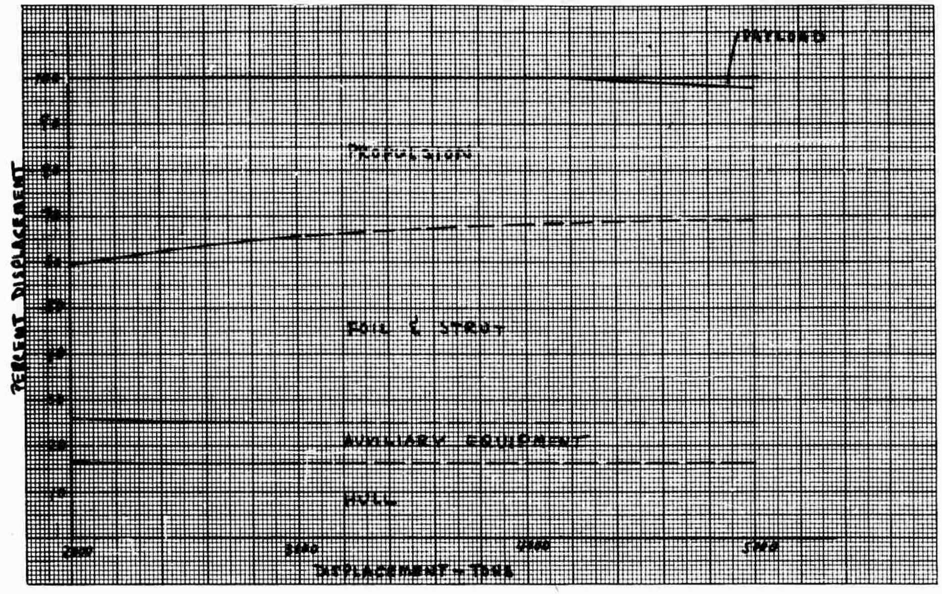
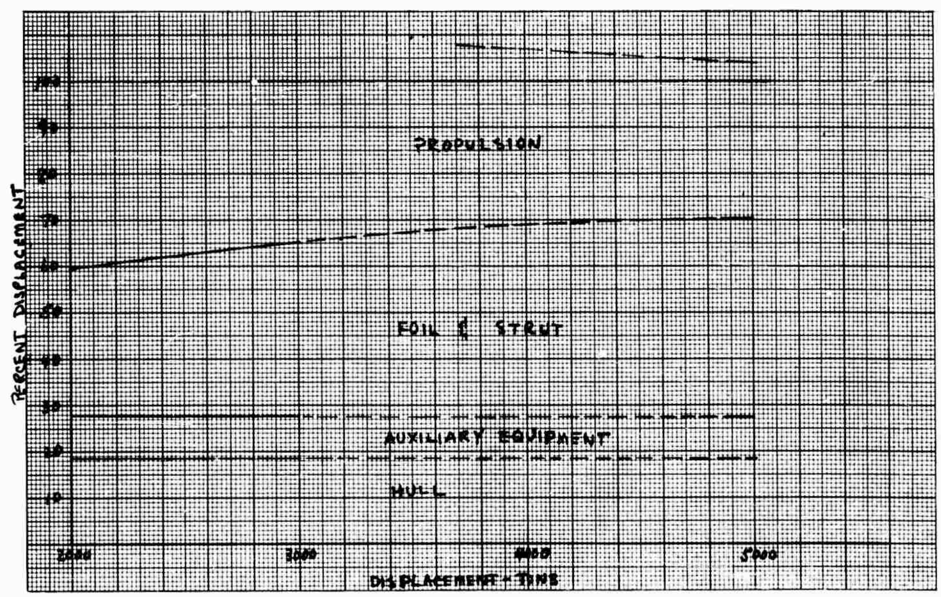


FIG. C-28 HYDROFOILS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE
OF FULL LOAD DISPLACEMENT, NUCLEAR POWER PLANT
(Speed, 60 Knots; Supercavitsting Foils)



Appendix D

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TABLES

ILLUSTRATIONS

Illustrations (continued)

Illustrations (continued)

Illustrations (continued)

Illustrations (concluded)

Appendix D

GEMS

Appendix D

GEMS

Introduction

The purpose of this study is to provide information for assessing the value of GEM's as a member of an amphibious fleet in the 1970-1980 time period.

The GEM is in its infancy. Further research effort and operating experience on existing GEM's are necessary to establish understanding of the problems of cruising performance, stability, control, dynamic stability over waves, bottom shape, the merits of possible jet arrangements, and other design features such as Westland's patented flexible skirts, and so forth. Hence, it is rather premature to predict the actual configuration of a GEM of several thousand tons displacement. On the other hand, judgments based on available information make it possible, and not as difficult, to predict the power to be installed and the relationship between payload, cruise velocity, and range.

GEM's chosen for this study are those having a jet arrangement consisting of a single continuous annular jet at the periphery of the craft.

List of Symbols

A	lift augmentation factor
a	width of the base in feet
b	length of the base in feet
C	perimeter of the base in feet measured at the centerline of nozzle
C_{Df}	parasite drag coefficient
C_L	lift coefficient $= \frac{L}{\frac{1}{2}\rho v_o^2 S}$
C_{LO}	aerodynamic lift coefficient
D	net drag in pounds
G	nozzle width in feet

h	height above surface in feet
HP_c	cushion horsepower
HP_p	propulsive horsepower
HP_T	total shaft horsepower installed
J	total annular jet momentum flux at nozzle exit in pounds
L	total lift (= gross weight) in pounds
M_j	total jet mass flow in slugs per second (including secondary jets)
q_o	freestream dynamic pressure in pounds per square foot
q_j	average jet dynamic pressure at nozzle exit in pounds per square foot
S	base area in square feet
S_{fan}	swept area of cushion fans in square feet
V_o	freestream velocity in knots
v_o	freestream velocity in ft/sec
v_j	average jet velocity in feet per second
β	tangential jet deflection angle in degrees
ΔP_{fan}	pressure rise across fan in pounds per square foot
η_A	augmentation efficiency
η_{fan}	fan efficiency
η_F	propulsive efficiency
θ	normal jet discharge angle in degrees
ρ	mass density of air in slugs per cubic foot
Δ	gross weight in tons

Estimation of Total Power Required

A relatively accurate prediction of hovering performance can be made within the current state of technology. However, there is no compatible information available for predicting cruising performance. For preliminary design purposes, a simple theory based on the following assumptions is generally used:

Lift = static lift + aerodynamic lift

Drag = parasite drag + momentum drag.

Precise estimates will not be possible until comprehensive experimental data are available. The following performance equations (for $\beta = o$), mainly derived from Reference No. 1, are based on the simple theory noted above and will be used in this study.

Lift: $$L = A\,J\,\left(1-0.104\,\frac{2}{C_L}\right) + C_{LO}\,q_o S \qquad (1)\ \text{(Ref. 1)}$$

where the lift augmentation factor, A, is given by:

$$A = \cos\theta + (1-\sin\theta)\,\frac{S}{h\,C}\,\eta_A\ . \qquad (2)\ \text{(Ref. 2)}$$

Since $J = 2q_j\,GC$, then,

$$q_j = \left(\frac{L}{2AGC}\right)\frac{\left(1-\dfrac{0.4}{C_L}\right)}{\left(1-\dfrac{0.208}{C_L}\right)}\ . \qquad (3)$$

The mass flow, M_j, is given by:

$$M_j = 1.2\ P\ G\ C\ v_j \qquad (4)$$

where the factor 1.2 accounts for the air to be fed to the stabilizing (secondary) nozzles. By using $q_j = \frac{1}{2}\rho v_j^2$, Eq. (4) can be written, in terms of q_j, as follows:

$$M_j = 1.7\sqrt{\rho}\,\sqrt{q_j}\ GC\ . \qquad (5)$$

The fan pressure rise, ΔP_{fan}, is calculated from:

$$\Delta P_{fan} = \left[1 + 1/2 \frac{(GC)^2}{S_{fan}^2}\right] q_j + \frac{L}{2S} \frac{(1-0.7)}{A} \frac{(1-0.4)}{C_L} - 0.9 q_o \qquad (6) \text{ (Ref. 1)}$$

where 90 percent ram efficiency is assumed.

The cushion horsepower, HP_c, is then

$$HP_c = \frac{M_j \; \Delta P_{fan}}{550 \; \rho \; \eta_{fan}} \qquad (7)$$

and drag is

$$D = M_j v_o + C_{Df} \; q_o \; S \quad . \qquad (8)$$

The propulsive horsepower, HP_p, is calculated from

$$HP_p = \frac{Dv_o}{550 \; \eta_p} \quad . \qquad (9)$$

The total power required is then given by:

$$HP_T = HP_c + HP_p \quad . \qquad (10)$$

Here, it is tacitly assumed that the cushion power and the propulsive power can be intertransferred in accord with the cruising conditions.

Chaplin[3] suggests the following limits for practical designs:

$$0.1 < \frac{G}{h} \eta_A < 0.3$$

$$\frac{S}{h_c} \eta_A > 2$$

$$-90° < \theta < -30°$$

$$\frac{L}{S} < 200 \text{ lbs/ft}^2 \quad .$$

For operation within this range, he also gives:

$$\eta_A = 0.8$$

$$\eta_{fan} = 0.8 - 0.9$$

$$\eta_p = 0.8 - 0.9 \quad .$$

The following values of parameters and efficiencies were selected or assumed for use in this study:

$$\frac{G}{h}\,\eta_A = 0.20$$

$$\theta_\beta = -45°$$

$$\beta = 0$$

$$\frac{L}{S} = 50 \text{ lbs/ft}^2$$

$$S_{fan} = 0.1S$$

$$\eta_A = 0.8$$

$$\eta_{fan} = 0.85$$

$$\eta_p = 0.85$$

$$C_{Df} = 0.07$$

$$C_{LO} = 0.4 \quad .$$

The plan form used is as shown in Figure D-1.

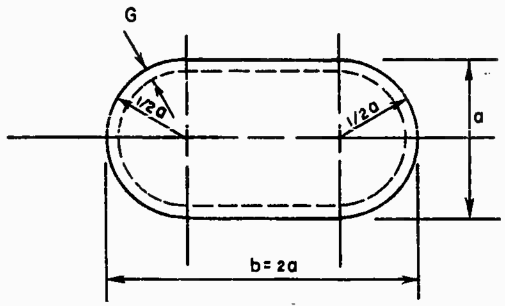


FIG. D-1 ASSUMED PLAN FORM

Table D-I shows the plan form dimensions.

The calculations were carried out for hovering heights of 8, 10, 12, 15, and 20 feet and cruising velocities of 60, 80, and 100 knots and the results shown in Figs. D-2, D-3, and D-4.

Vehicle Weight Estimates

Power Plant Weight, W_{pr}

Power plants using an aircraft-type, fuel-powered gas turbine prime mover and a closed-cycle, gas-cooled nuclear power plant were considered.

Power Plant with Gas Turbine Prime Mover

The weight breakdown for the power plant of Vickers VA-3 (L = 28,400 lbs) is as follows:[4]

Item	Weight (lb)	Weight/HP
Engines, 4 x 425 HP free-turbine	2,000	1.18
Fans, propellers, gear boxes	3,200	1.88
Total	5,200	3.16

Note: No interconnection between lift and propulsion engines.

Table D-I

PLAN FORM DIMENSIONS VERSUS TOTAL DISPLACEMENT

Δ (tons)	L(10^6lbs)	S(10^3ft^2)	a (ft)	b (ft)
500	1.12	22.4	78.2	156.4
1,000	2.24	44.8	112	224
2,000	4.48	89.6	158.5	317
3,000	6.72	134.4	194	388
4,000	8.96	179.2	224	448
5,000	11.2	224	251	502
6,000	13.44	268.8	274	548
7,000	15.68	313.6	296	592
8,000	17.92	358.4	317	634
9,000	20.16	403.2	336	672
10,000	22.4	448	354	708

It is noted that, if a means is provided to interconnect lift and propulsion engines, the weight for the transmission should be added to the above. However, if an aircraft-type gas turbine in the order of 0.6 lb/HP is used as an engine, some weight would be saved. It was found that, in most cases, the power plant weight is a rather small fraction of the gross weight of the vehicle. Hence a specific weight of 3 lbs/HP was chosen in this study to allow for the use of some heavy components in the interest of cost savings.

This value is certainly on the conservative side when compared to 1.1 lbs/HP as used in Reference No. 5 (p.104).

Nuclear Power Plant

If a lightweight nuclear power plant, as discussed in Reference Nos. 6 and 7, could be developed in time, it would be quite attractive for large-size vehicles with long range operating capabilities. The weight of a nuclear power plant is strongly affected by the shielding assumptions. For example, for a 30,000-HP plant, Reference No. 6 gives 16 lbs/HP, but Reference No. 7 gives, on the other hand, 39 lbs/HP. Both are based on a closed-cycle, gas-cooled reactor located above water. The power plant weight/horsepower ratio for powers up to 200,000 HP given in Reference No. 7 can be calculated by:

$$\text{lbs/HP} = \frac{282}{(\text{HP} \times 10^{-3})^{0.582}} \quad . \tag{11}$$

This ratio as a function of total horsepower is shown in the lower curve of Fig. D-5.

For this study it was decided that data from the Grumman study[7] should be adopted. But there is a problem as to how the Grumman study should be extrapolated to the order of 1,000,000 horsepower. It is a well-known fact, as can be seen from Eq. (11), that the nuclear reactor and its shielding, and hence the total plant weight per horsepower, decrease rapidly as the power increases. Hence, for the power plant in question, the plant weight per horsepower may also be predicted by a formula of the form:

$$\text{lbs/HP} = \frac{K_1}{(\text{HP})^{K_2}} \quad .$$

where K_1 and K_2 are some constants to be supplied from reliable data. Since there are no reliable data available, Eq. (11) was arbitrarily used for the power range in this study. At 1,000,000 HP, the equation gives 5 lbs/HP.

To Eq. (11), 2 lbs/HP was added to account for weights of fans, transmissions, and propellers, that is, the total power plant/horsepower was given by:

$$\text{lbs/HP} = \frac{282}{(\text{HP} \times 10^{-3})^{0.582}} + 2 \quad . \tag{12}$$

Equation (12) is also plotted in Fig. D-5.

Structure Weight, W_s

The ONR study[5] (for amphibious support crafts) shows that the structural weight of a GEM is well defined by:

$$\frac{W_s}{S} = 5 + 0.175 \frac{L}{S} \quad (\text{lbs/ft}^2) \quad . \tag{13}$$

At L/S = 50 lbs/ft^2, this equation gives W_s = 13.75 S lbs.

It follows that

$$\frac{W_s}{\Delta} = \frac{13.75\ S}{50S} = 27.5 \text{ percent.}$$

The Ryan study[8] gives $\frac{W_s}{\Delta}$ = 26.4 percent for a GEM of 40 tons displacement at L/S = 48.5 lbs/ft^2. Reference No. 6 gives $\frac{W_s}{\Delta}$ = 17 percent for a GEM of Δ = 306 tons at L/S = 50 lbs/ft^2.

In this study, 25 percent of the displacement has been allowed for the weight of the structure, using aluminum alloys.

Outfit Weight, W_o

Ten percent of displacement was assigned to the outfit weight.

Fuel Weight, W_F

Breguet's range equation, Eq. (14), was used to estimate specific fuel weight, W_F/Δ; thus,

$$\frac{W_F}{\Delta} = 1 - \frac{1}{e^x} \tag{14}$$

where:

$$x = \frac{R\ (SFC)\ HP_T}{L\ V_o}$$

R = range in nautical miles

SFC = specific fuel consumption on lbs/HP/hr.

Payload, W_{PL}

The payload is given by:

$$W_{PL} = \Delta - (W_{pr} + W_s + W_o + W_F) .$$

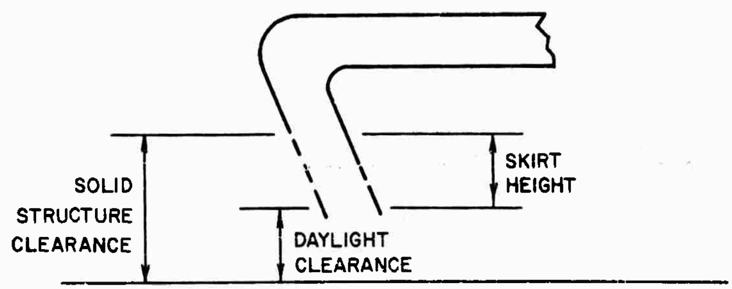
Summary Curves

The results of the weight estimates are summarized in Figs. D-6 through D-41, from which an appraisal of GEM's can be made. Curves for both nuclear and gas-turbine propulsion are shown. All curves assume L/S = 50 lbs/ft^2.

Westland's Flexible Skirt

Significant design progress has recently been made by Westland on flexible skirts. Some extracts from the article "Westland Ride High" (May 1963 issue of Reference No. 4) are noted below:

> One of the major factors that has contributed to the success of the Westland range of air cushion vehicles is a device known as the Westland patented skirt . . . Terms used in discussing hovercraft



skirts are the distance from the hemline to the solid structure--which is called the skirt height--and the distance from hemline to the ground, which is called the daylight clearance, and which determines the power required to generate the air cushion . . . These skirts were so successful that they enabled SR N2 to operate in 6 ft sea with a daylight clearance of only 9 inches . . . When operating over water without skirts, severe impacts occur when the wave height becomes much greater than the daylight clearance. When skirts are fitted the over-wave performance is increased by an amount almost equal to the length of the skirts . . . Four-foot skirts are by no means the practical limit of development. This is a model of Westland's new project for a 170 ton ferry - SR N4. This craft has 8 ft skirts and is operating under conditions simulating a speed of 70 knots and 7 ft waves.

Speed Keeping Ability of GEM in Waves

The operating experience of the SR-N2 (Reference No. 4, September 1962), which was designed for a cruise speed of 70 knots in smooth water at h = 1 ft, shows that the craft can operate in 3- to 4-ft seas and winds up to 30 mph. Introducing the MK2 version of SR-N2, Westland remarked (Reference No. 4, September 1962):

> The effect on the craft of wave height depends on the length of the wave and the direction of approach, i.e., whether the craft is running into, down or across wind. In general the longer the sea, the higher it can be. Little or no reduction of speed is necessary in seas up to 1-1/2 feet high. In waves of critical length (approximately twice craft length), SR-N2 MK2 will operate at around 45 knots in waves 3 ft to 4 ft high and can handle much larger waves at lower speeds.

This shows that SR-N2 can operate in a wave height of 1.5h without reduction of speed.

Accordingly, we may speculate that a GEM designed to operate at h = 10 ft could operate, with the use of skirts, or some other means yet to be developed, in state 4-5 seas at designed cruise speed, and a GEM of h = 15 ft in state 5-6 seas. It should be pointed out that, if in a rough sea, a GEM is required to operate at a reduced cruise speed, Va, then the range, R, in the summary curves should be modified by:

$$Ra = Ro \left(\frac{Va}{Vo}\right) \text{ nautical miles}$$

where:

Ra = actual range in nautical miles

Ro = designed range in nautical miles

Va = actual average cruise velocity in knots

Vo = designed cruise velocity in knots.

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1. Chaplin, H.R., Design Study of a 29-Foot GEM, DTMB Report 1521, Aero Report 999, April 1961

2. Loos, J.E., Symposium on Ground Effect Phenomena, Princeton University, p.303-324, October 1959

3. Chaplin, H.R., Symposium on Ground Effect Phenomena, Princeton University, p.57-86, October 1959

4. Air-Cushion Vehicle, Iliffe Transport Publications, Ltd., July 1962 to August 1963 (monthly magazine)

5. A Study of the Operational Feasibility of the Ground Effect Machine in the Amphibious Support Mission, ONR Report ACR/NAR-26, November 1962

6. Westmoreland, J.C., J.B. Dee, and J.E. Loos, A Conceptual Nuclear Propulsion System for Ground Effect Machines, IAS Paper No. 61-46, Institute of the Aerospace Sciences, January 1961

7. Grumman Aircraft Engineering Corporation, Study of Hydrofoil Seacraft, Maritime Administration, October 1958

8. Ryan Aeronautical Company, Aerospace Division, Structural Design Study of a Ground Effect Machine for Amphibious Support, Final Report, Report No. 62B009, Office of Naval Research, February 1962

FIG. D-2 GEMS
HORSEPOWER VERSUS TOTAL DISPLACEMENT
(Various Heights; Speed, 60 Knots)

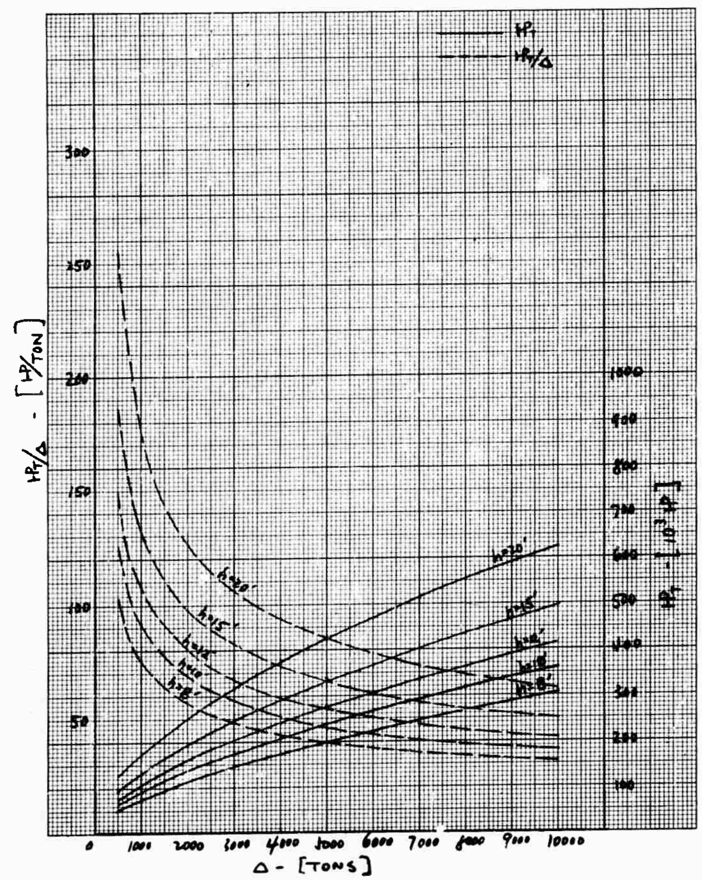


FIG. D-3 GEMS
HORSEPOWER VERSUS TOTAL DISPLACEMENT
(Various Heights; Speed, 80 Knots)

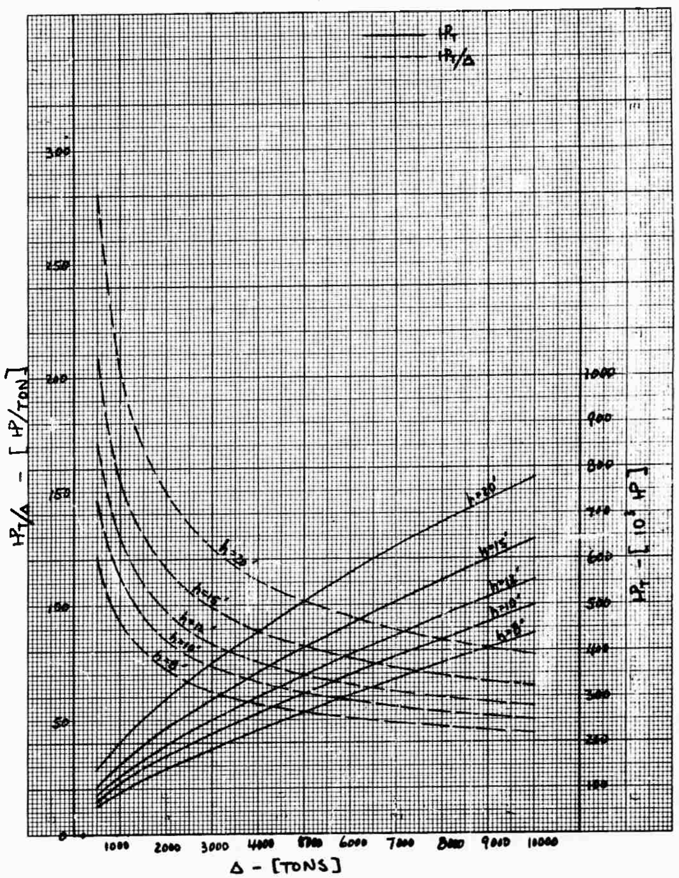


FIG. D-4 GEMS
HORSEPOWER VERSUS TOTAL DISPLACEMENT
(Various Heights; Speed, 100 Knots)

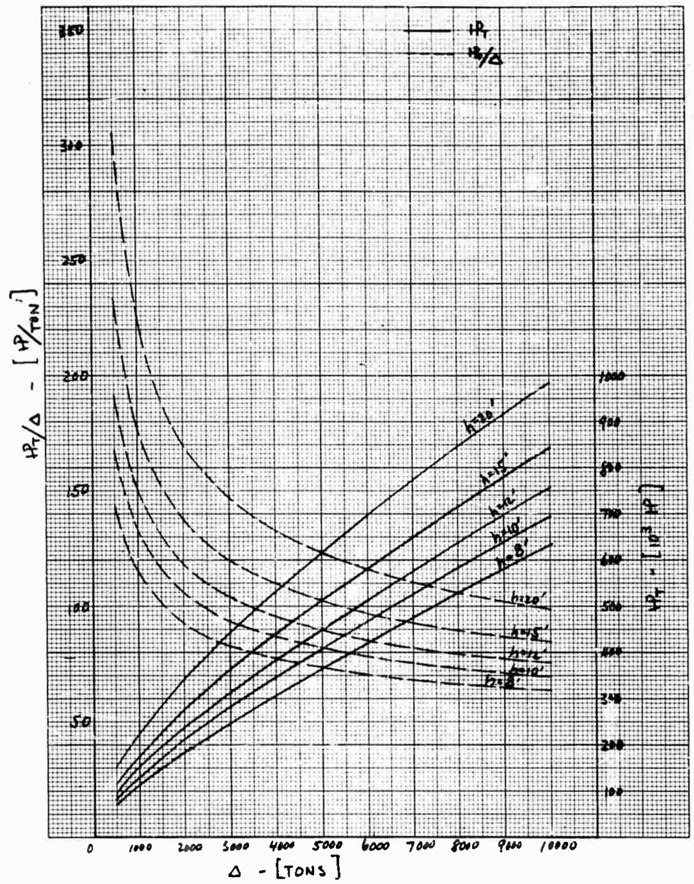
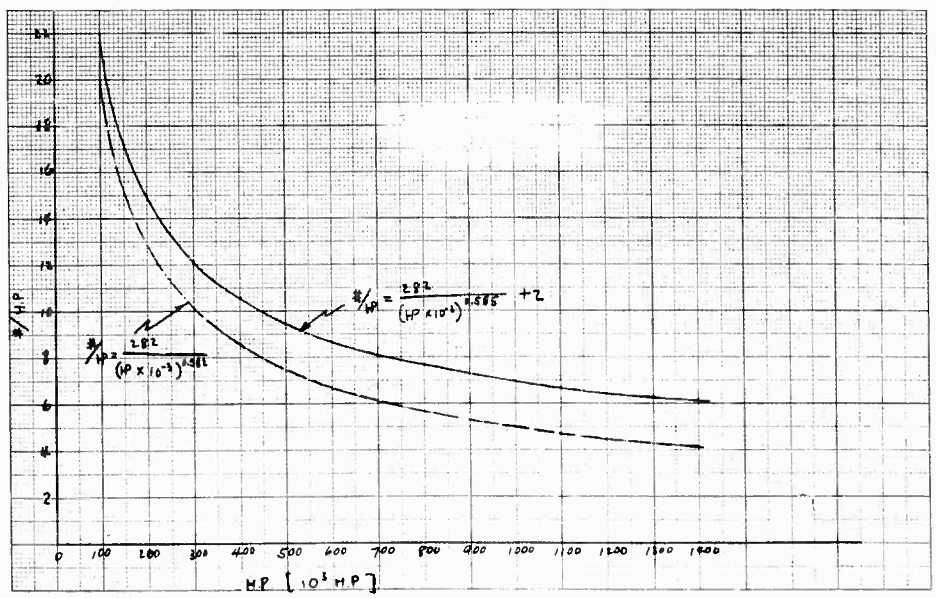


FIG. D-5 GEMS
POUNDS PER HORSEPOWER VERSUS HORSEPOWER
(Closed-Cycle, Gas-Cooled Nuclear Reactor)



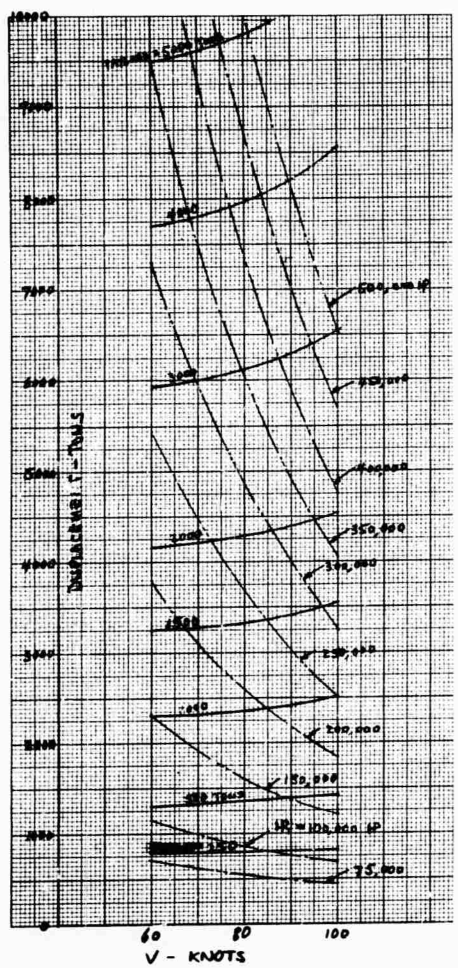


FIG. D-6 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 10 Feet;
Range, 500 Nautical Miles)

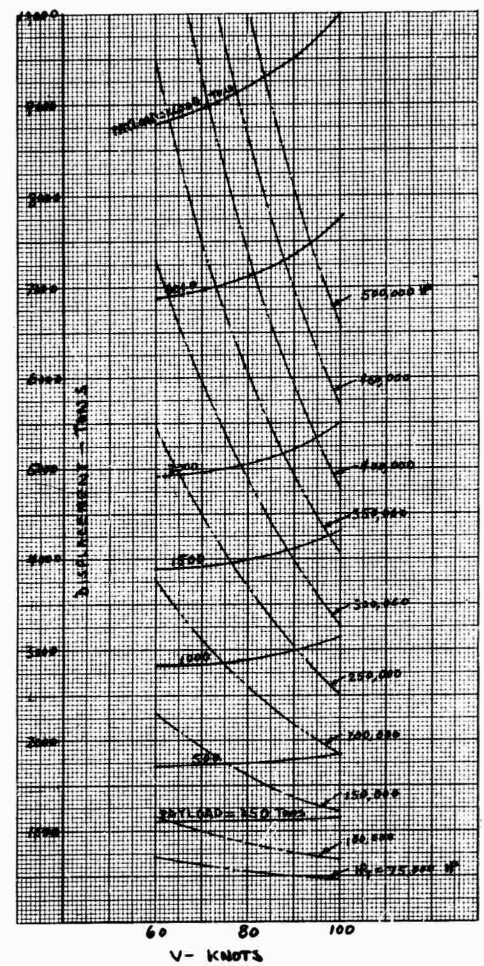


FIG. D-7 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 10 Feet;
Range, 1,000 Nautical Miles)

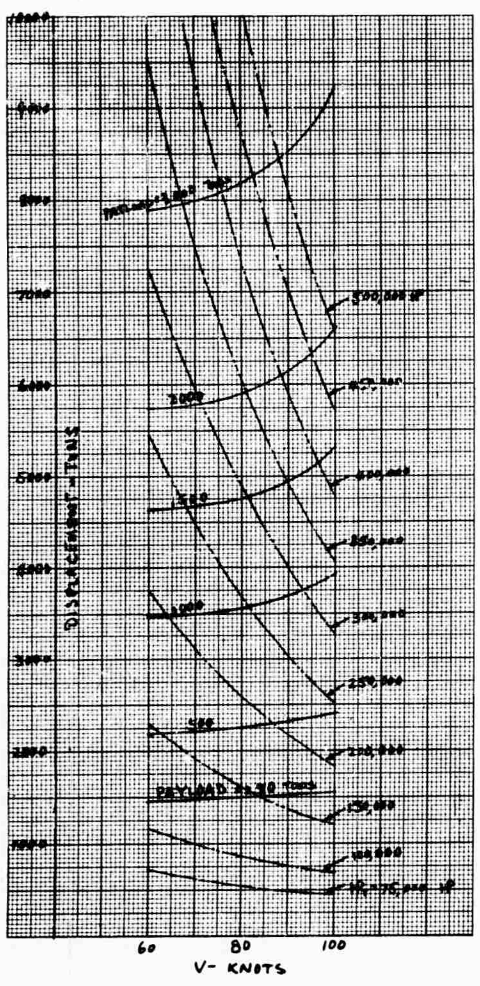


FIG. D-8 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 10 Feet;
Range, 1,500 Nautical Miles)

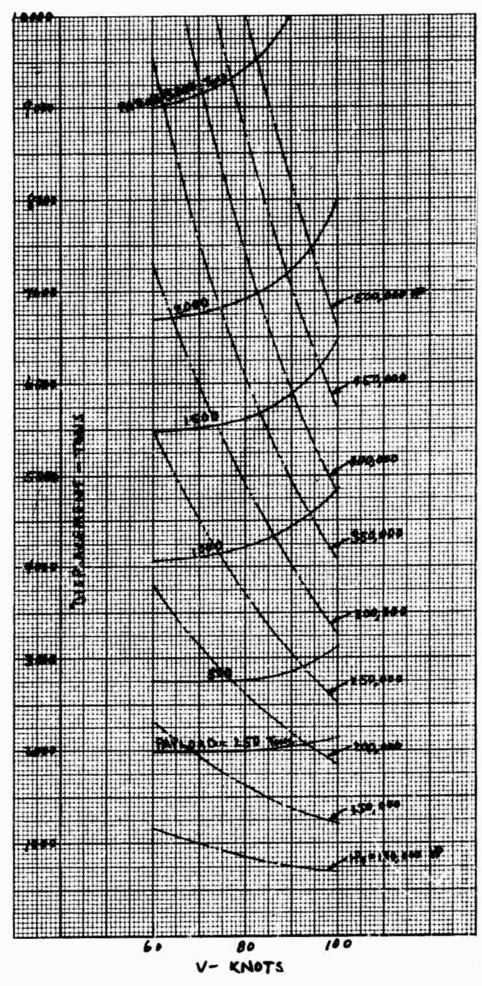


FIG. D-9 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 10 Feet;
Range, 2,000 Nautical Miles)

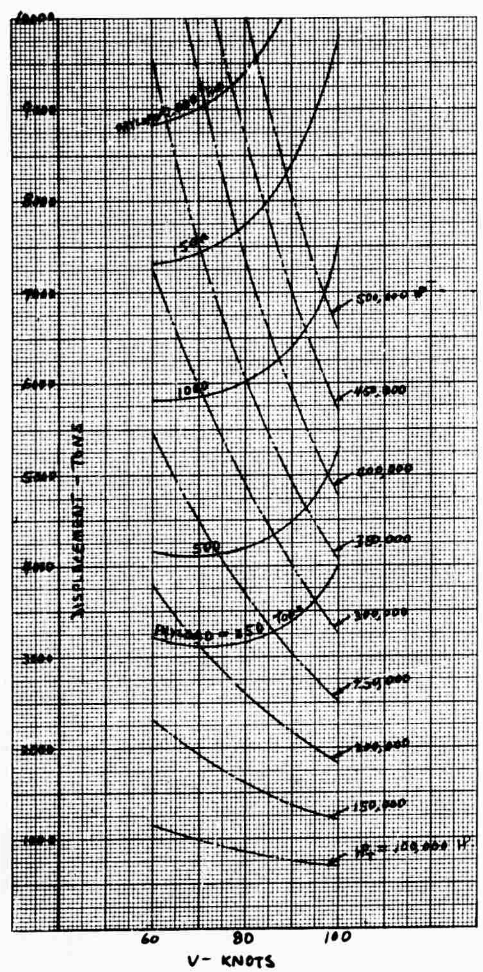


FIG. D-10 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 10 Feet;
Range, 3,000 Nautical Miles)

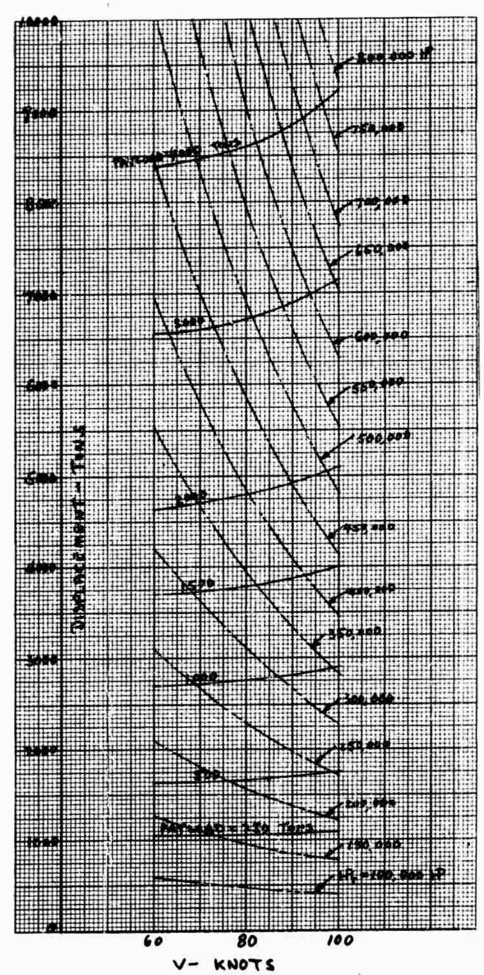


FIG. D-11 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 15 Feet;
Range, 500 Nautical Miles)

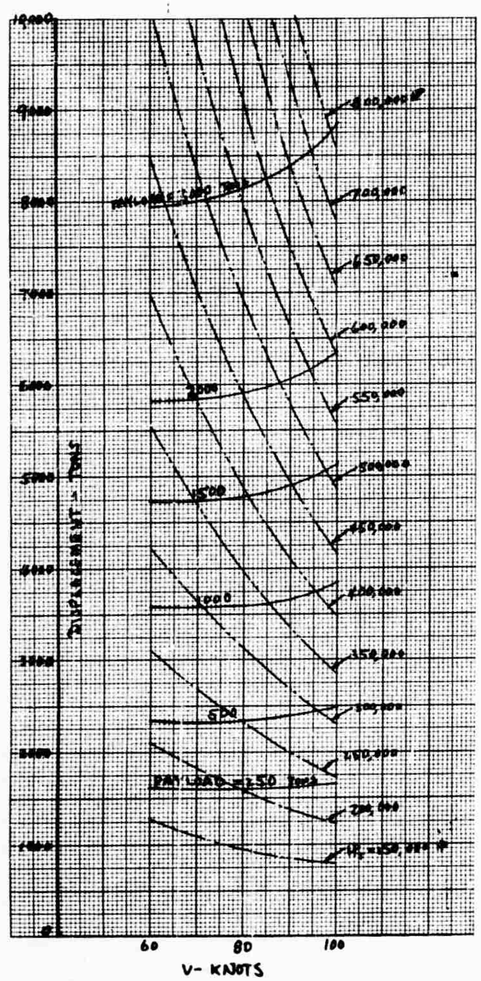


FIG. D-12 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 15 Feet;
Range, 1,000 Nautical Miles)

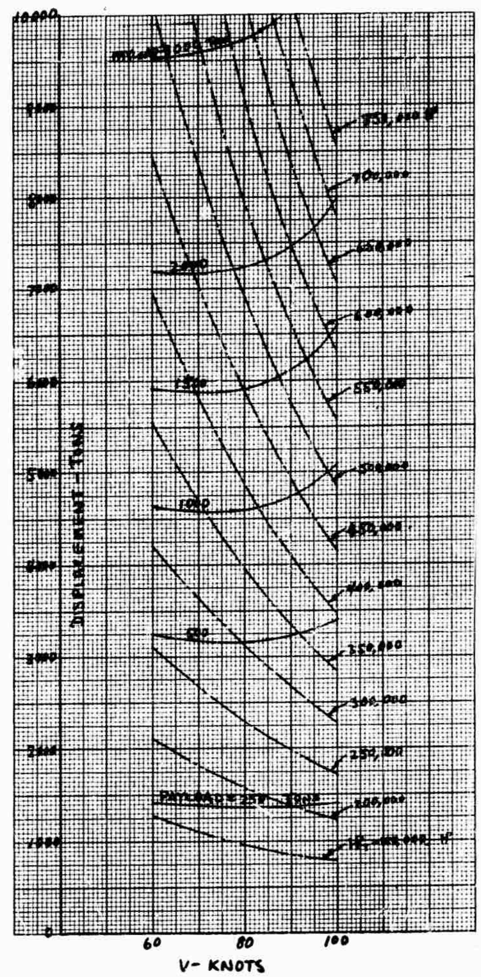


FIG. D-13 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 15 Feet;
Range, 1,500 Nautical Miles)

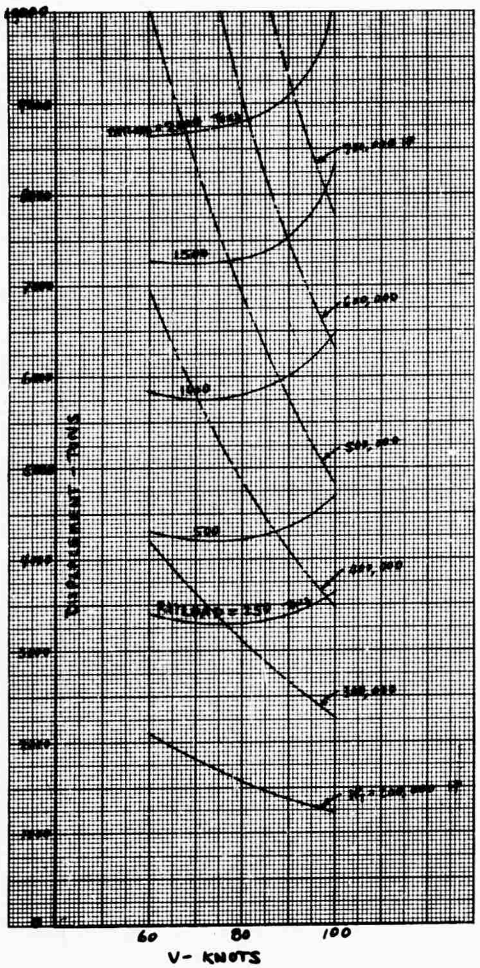


FIG. D-14 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 15 Feet;
Range, 2,000 Nautical Miles)

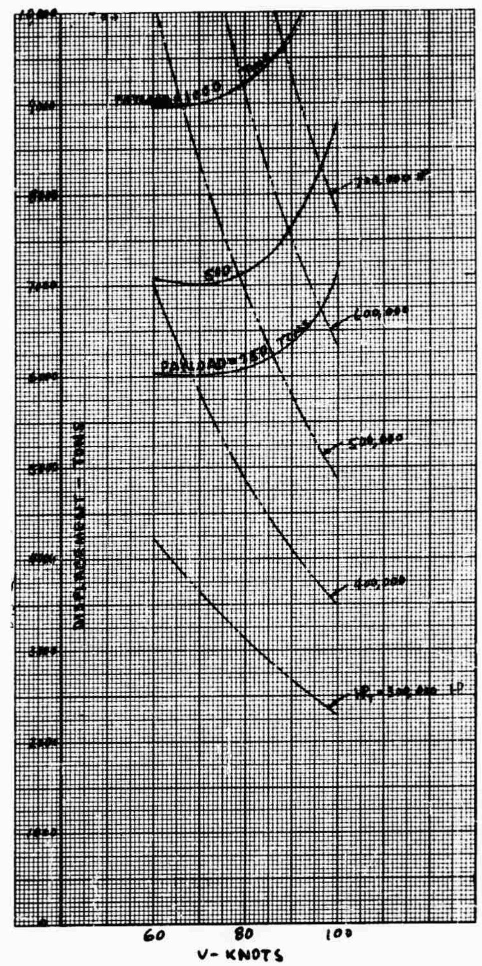


FIG. D-15 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 15 Feet;
Range, 3,000 Nautical Miles)

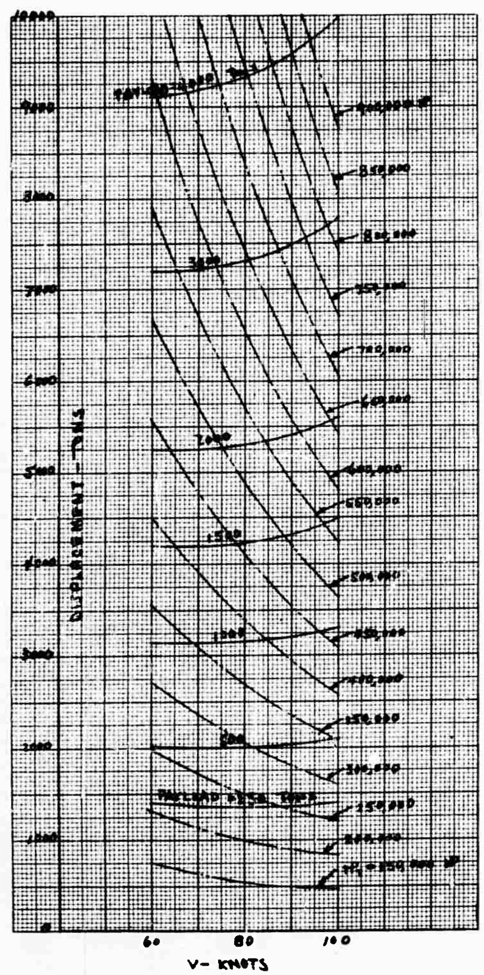


FIG. D-16 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 20 Feet;
Range, 500 Nautical Miles)

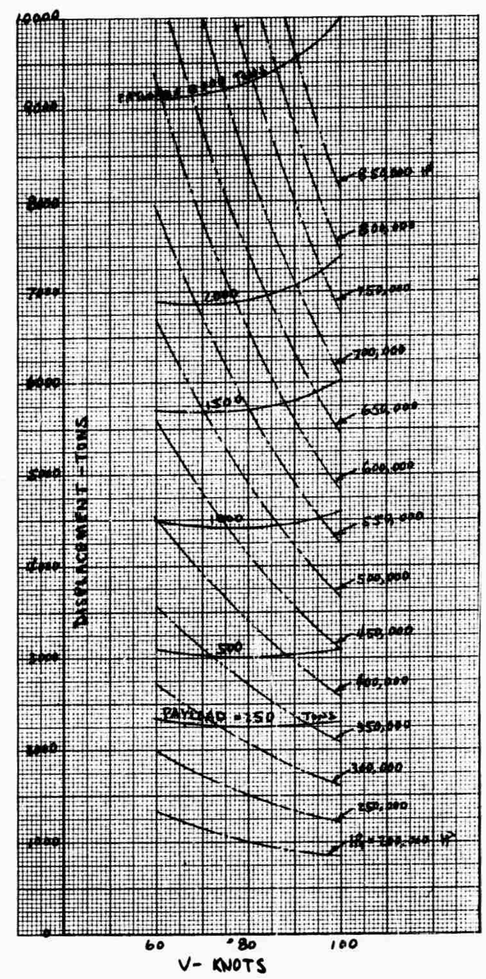


FIG. D-17 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 20 Feet;
Range, 1,000 Nautical Miles)

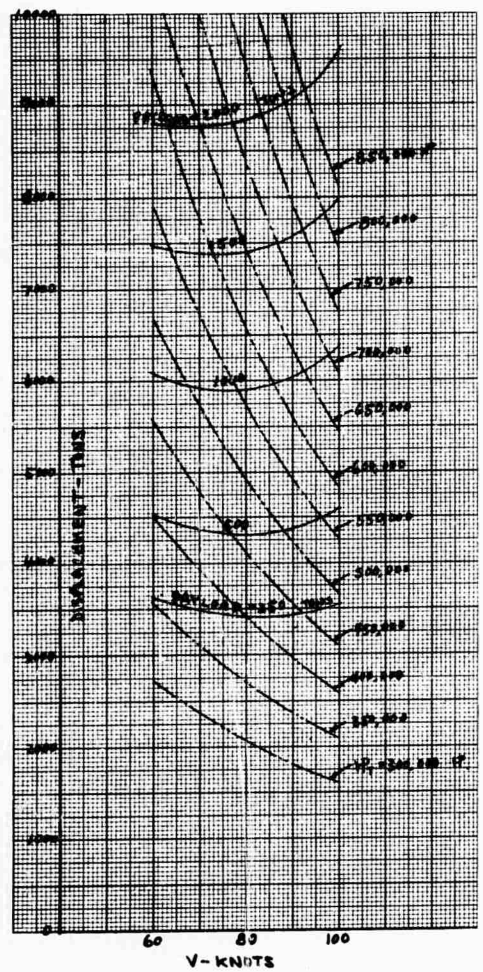


FIG. D-18 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 20 Feet;
Range, 1,500 Nautical Miles)

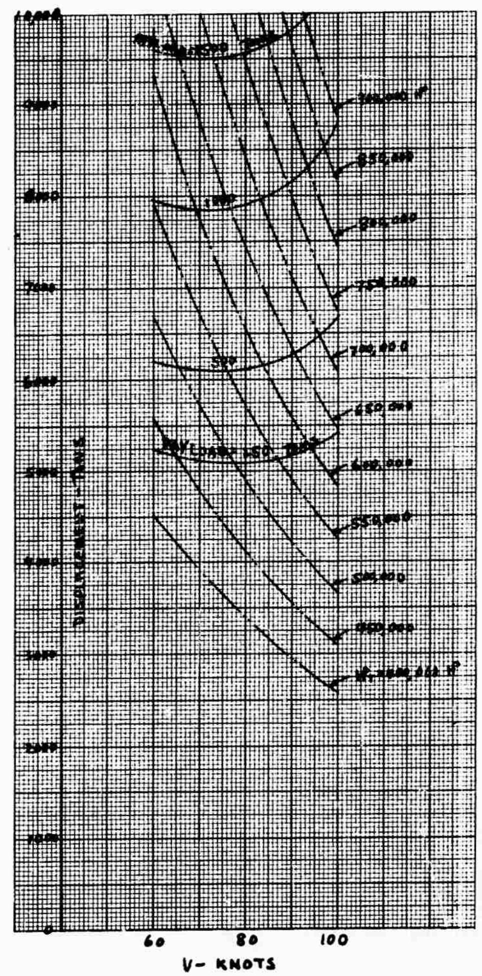


FIG. D-19 GEMS
DISPLACEMENT AND PAYLOAD VERSUS
SPEED AND REQUIRED SHAFT
HORSEPOWER, GAS TURBINE POWER PLANT
(Operating Height, 20 Feet;
Range, 2,000 Nautical Miles)

FIG. D-20 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 8 Feet; Speed, 60 Knots; Various Ranges)

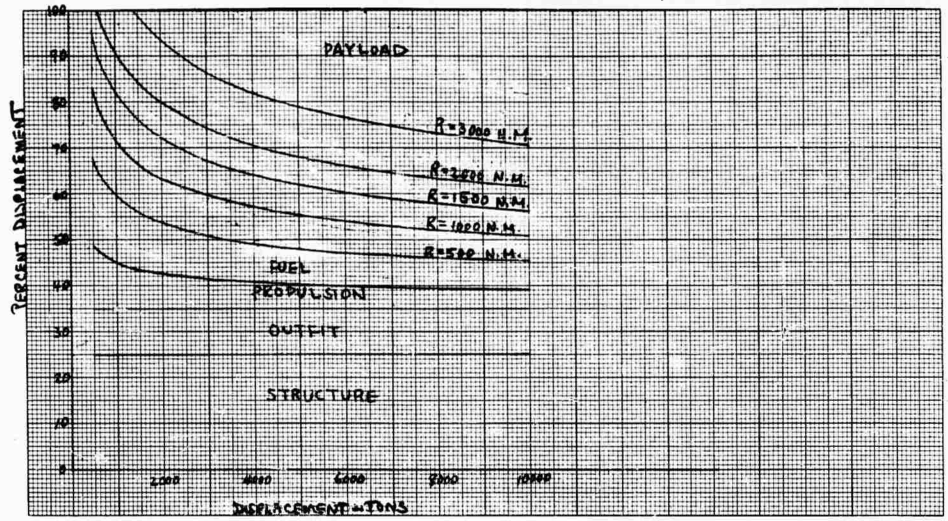


FIG. D-21 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 10 Feet; Speed, 60 Knots; Various Ranges)

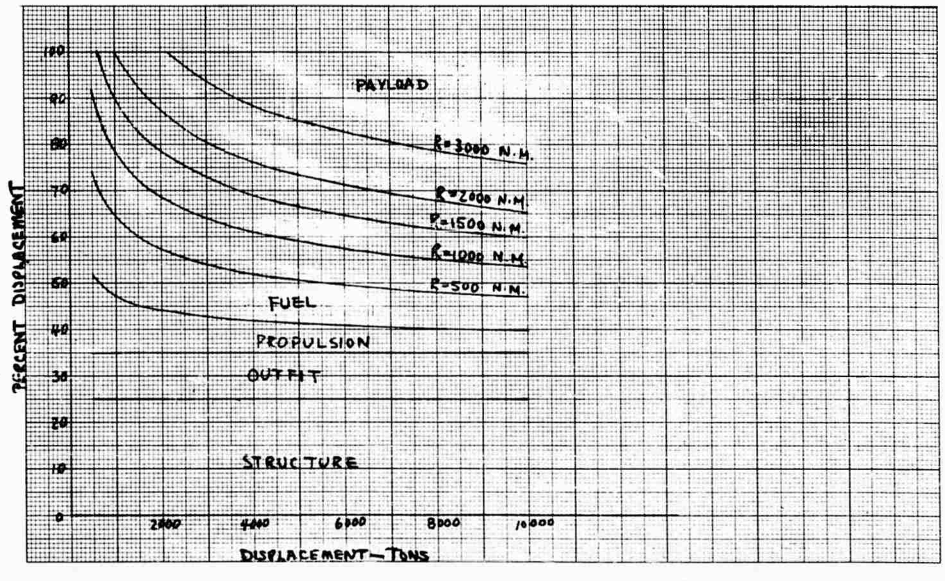


FIG. D-22 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD OISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 12 Feet; Speed, 60 Knots; Various Ranges)

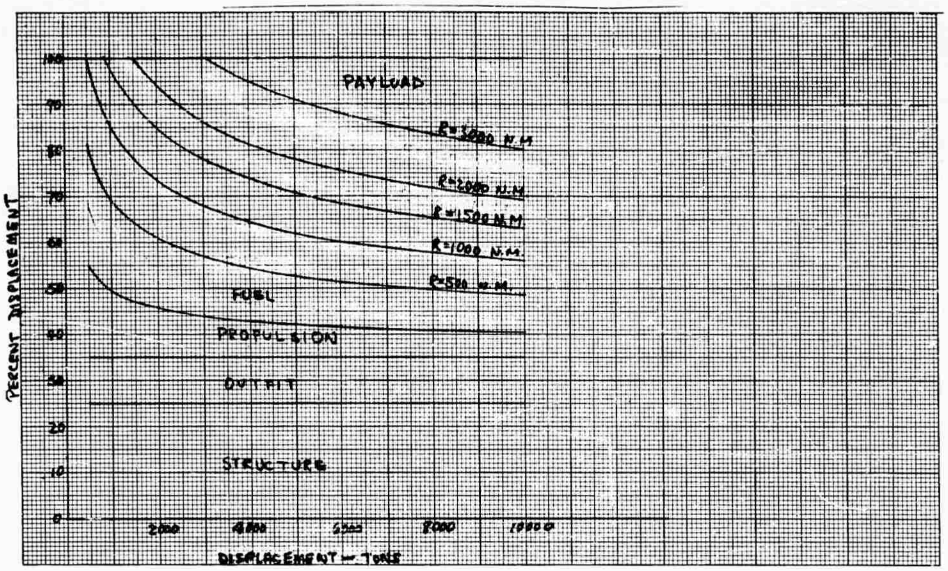


FIG. D-23 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD OISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 15 Feet; Speed, 60 Knots; Various Ranges)

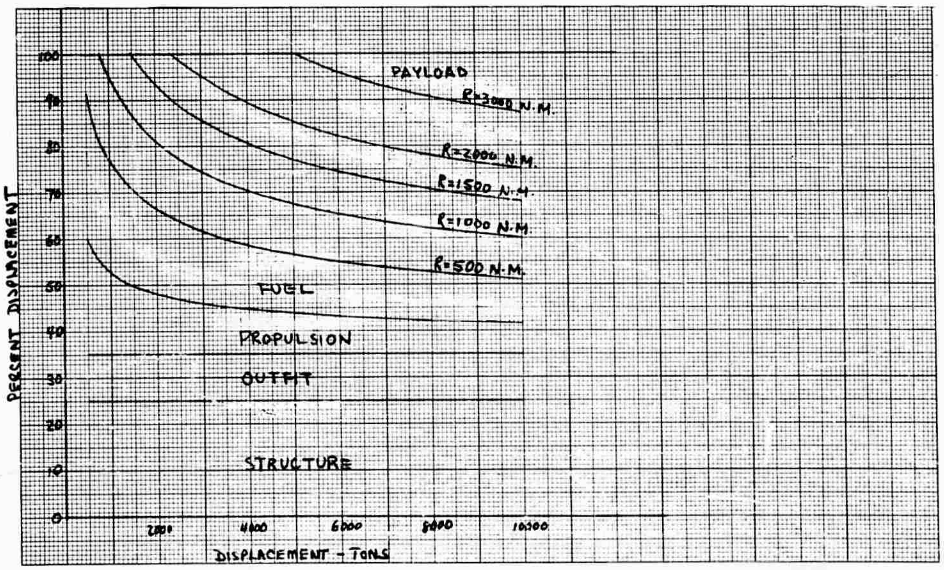


FIG. D-24 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 20 Feet; Speed, 60 Knots; Various Ranges)

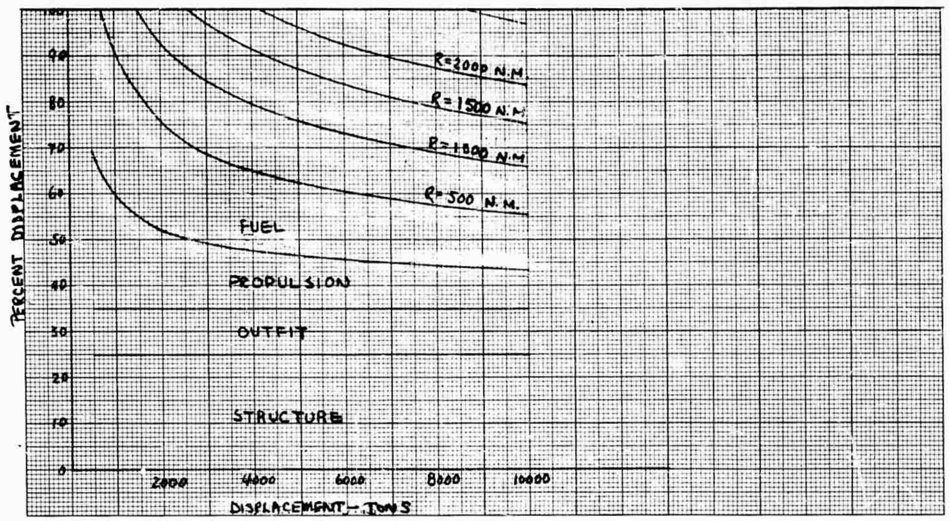


FIG. D-25 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 8 Feet; Speed, 80 Knots; Various Ranges)

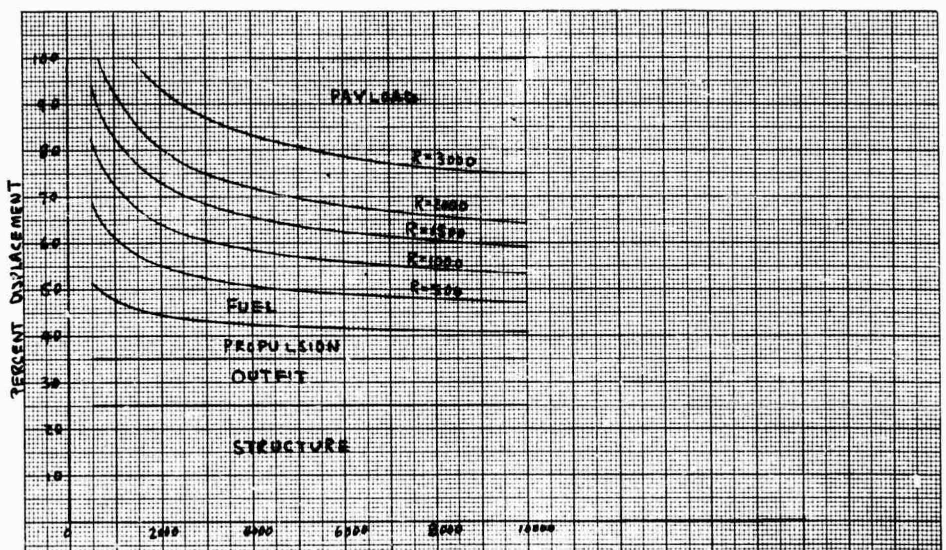


FIG. D-26 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 10 Feet; Speed, 80 Knots; Various Ranges)

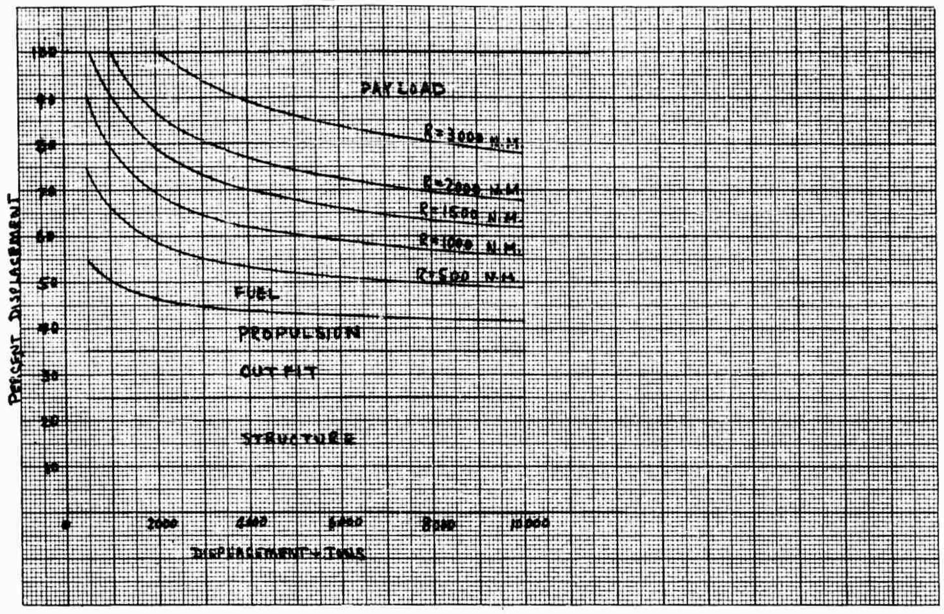


FIG. D-27 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 12 Feet; Speed, 80 Knots; Various Ranges)

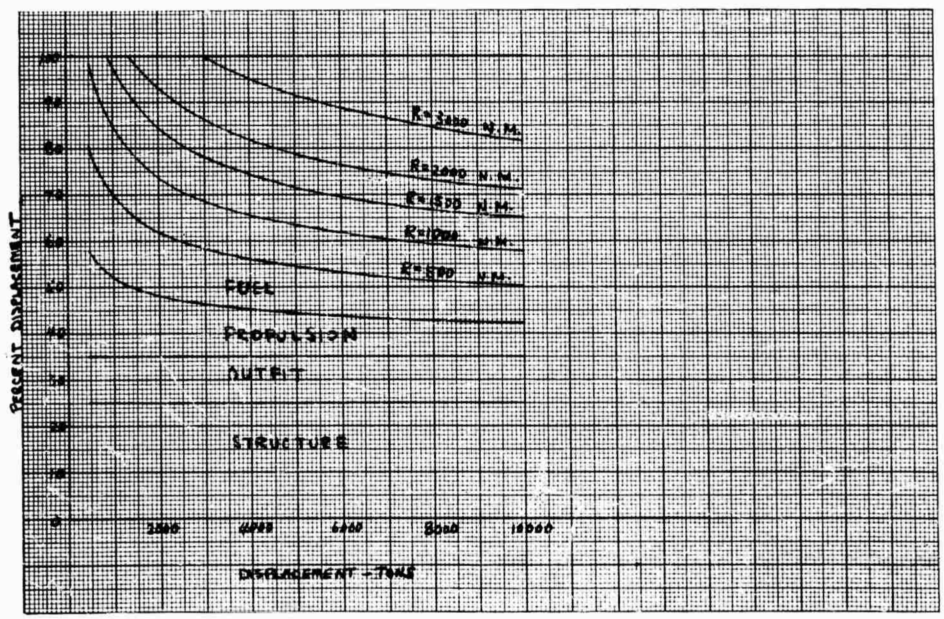


FIG. D-28 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 15 Feet; Speed, 80 Knots; Various Ranges)

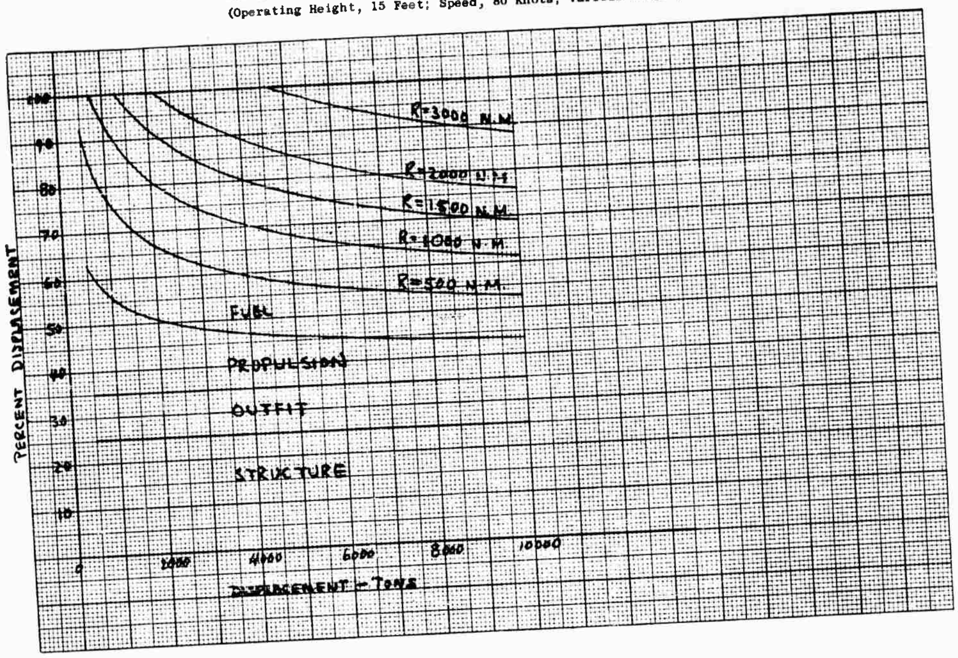


FIG. D-29 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 20 Feet; Speed, 80 Knots; Various Ranges)

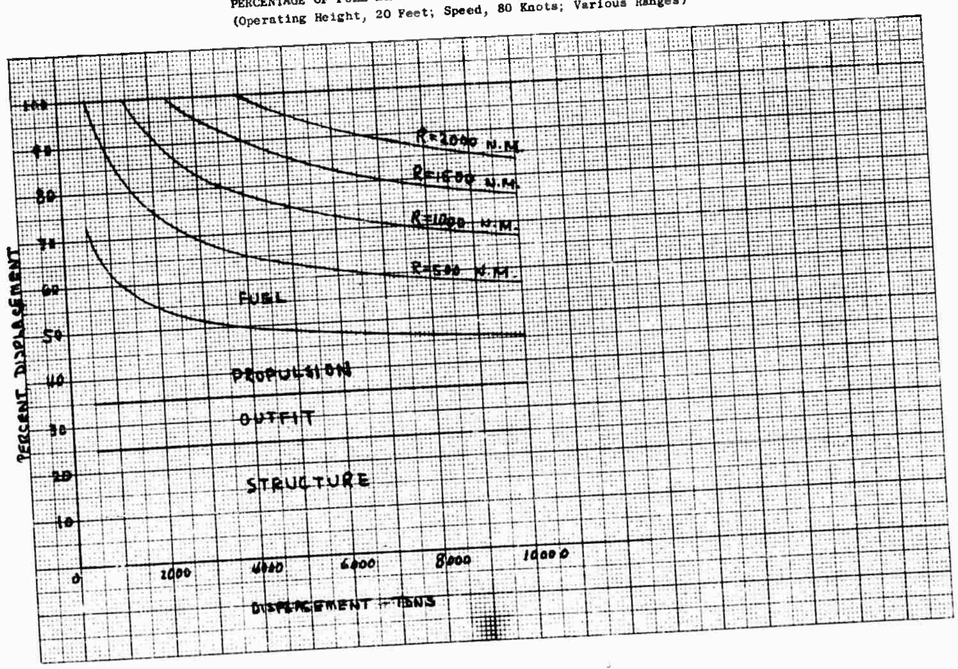


FIG. D-30 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 8 Feet; Speed, 100 Knots; Various Ranges)

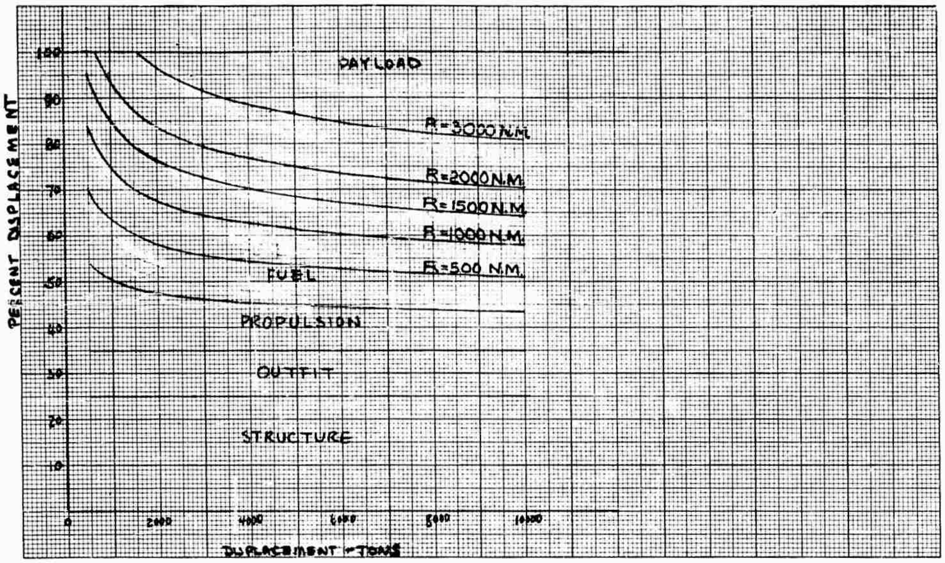


FIG. D-31 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 10 Feet; Speed, 100 Knots; Various Ranges)

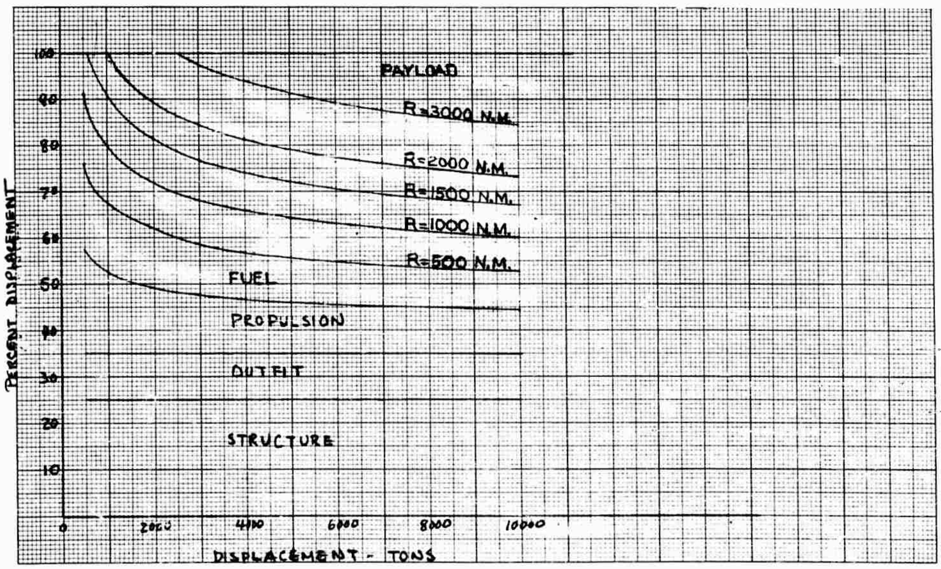


FIG. D-32 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 12 Feet; Speed, 100 Knots; Various Ranges)

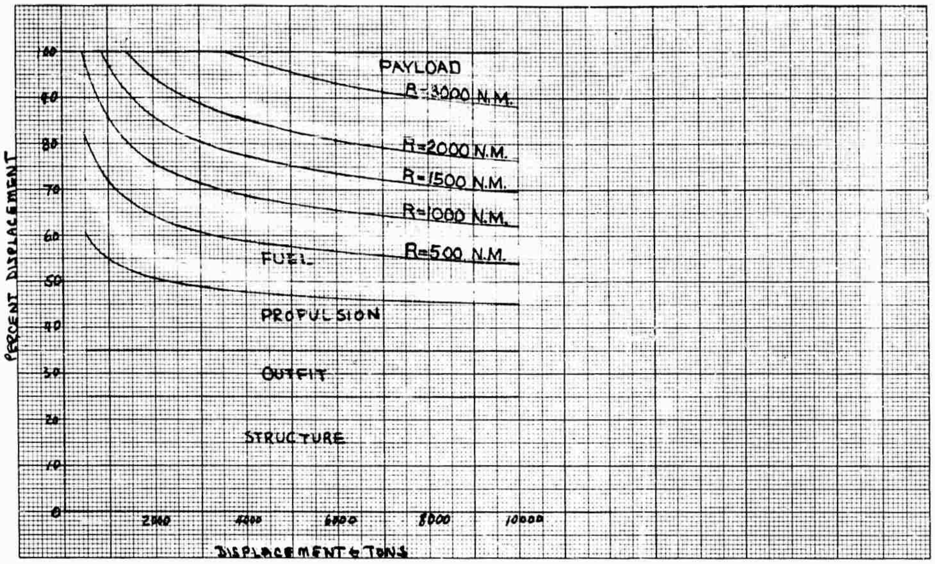


FIG. D-33 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 15 Feet; Speed, 100 Knots; Various Ranges)

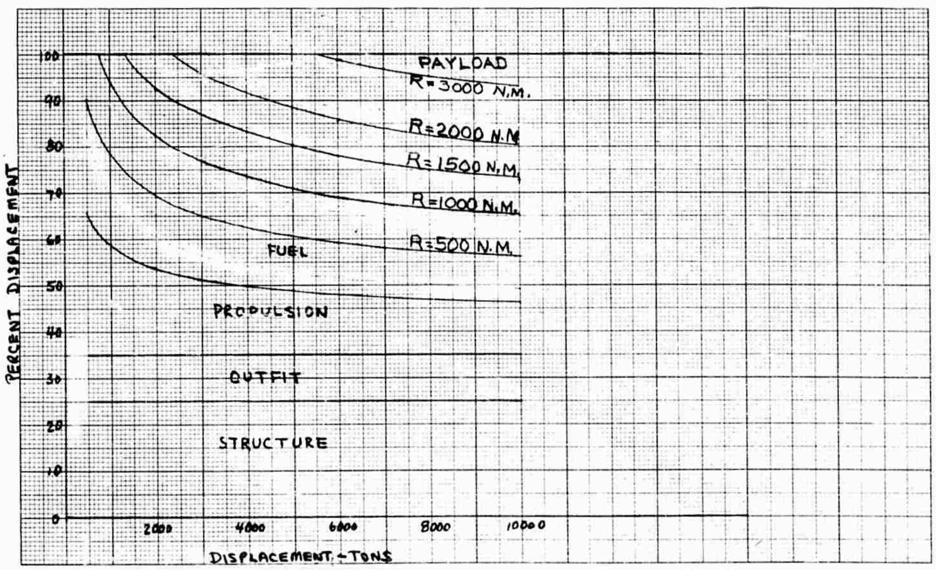


FIG. D-34 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, GAS TURBINE POWER PLANT
(Operating Height, 20 Feet; Speed, 100 Knots; Various Ranges)

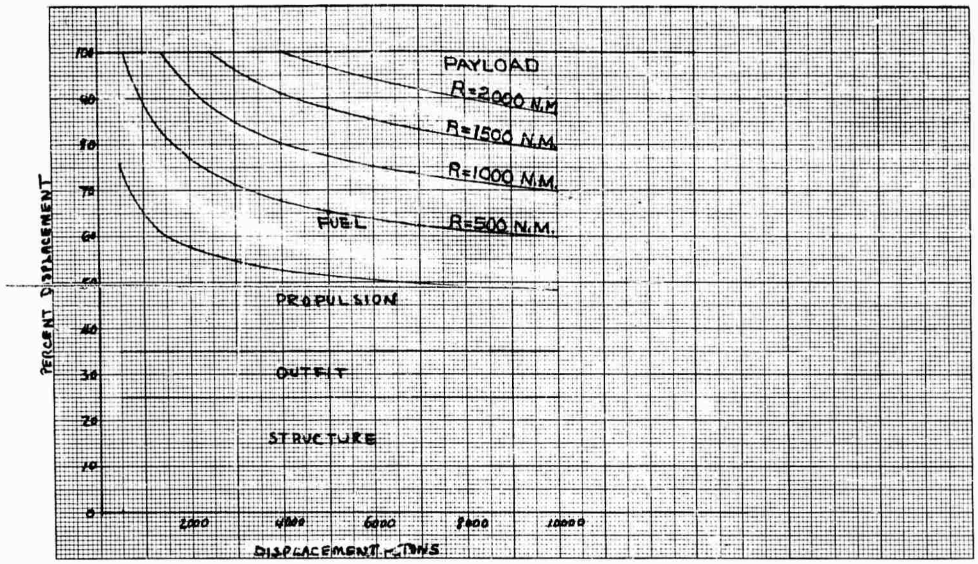


FIG. D-35 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE OF FULL LOAD DISPLACEMENT, NUCLEAR POWER PLANT
(Various Operating Heights; Speed, 60 Knots)

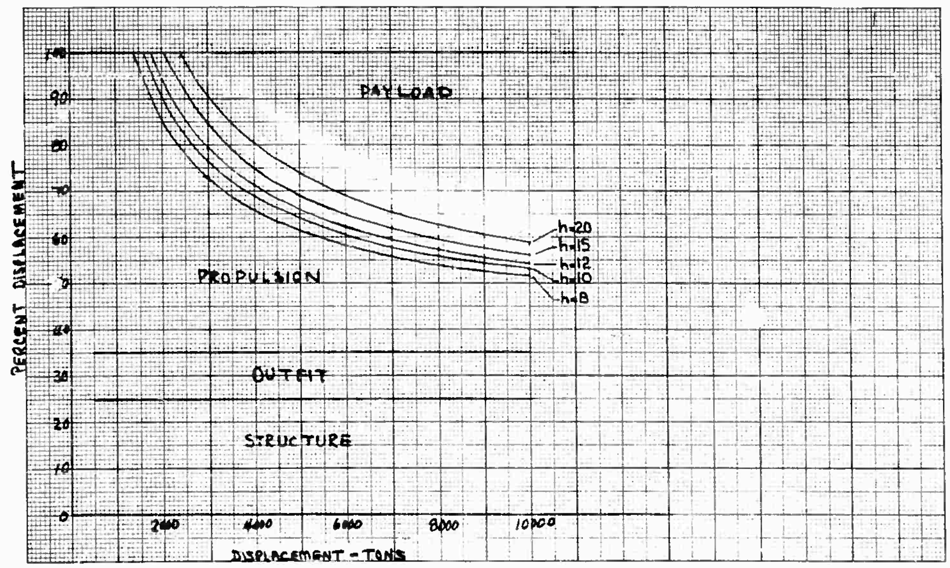


FIG. D-36 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT, NUCLEAR POWER PLANT
(Various Operating Heights; Speed, 80 Knots)

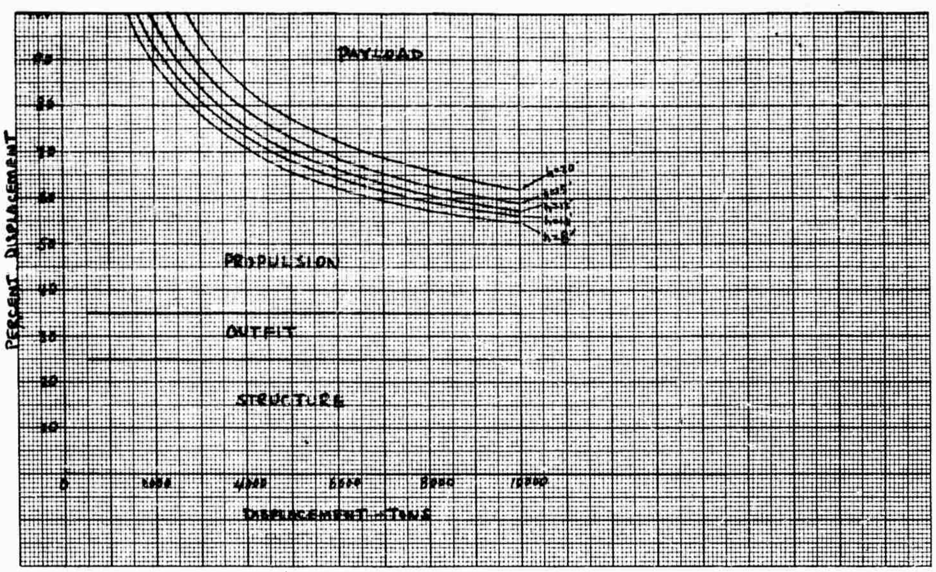


FIG. D-37 GEMS
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS
PERCENTAGE OF FULL LOAD DISPLACEMENT, NUCLEAR POWER PLANT
(Various Operating Heights; Speed, 100 Knots)

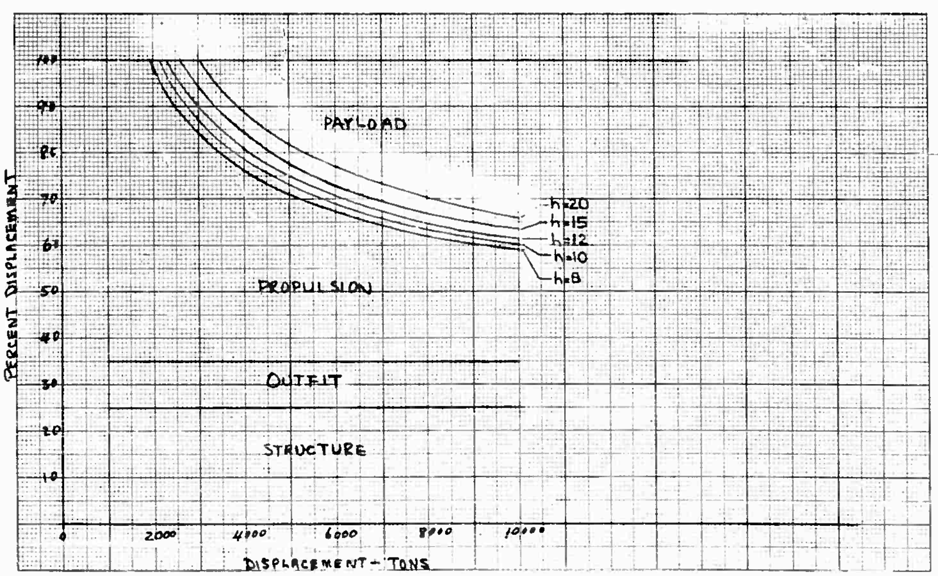


FIG. D-38 GEMS
PAYLOAD VERSUS RANGE, GAS TURBINE POWER PLANT
(Operating Height, 10 Feet;
Speed, 60, 80, and 100 Knots)

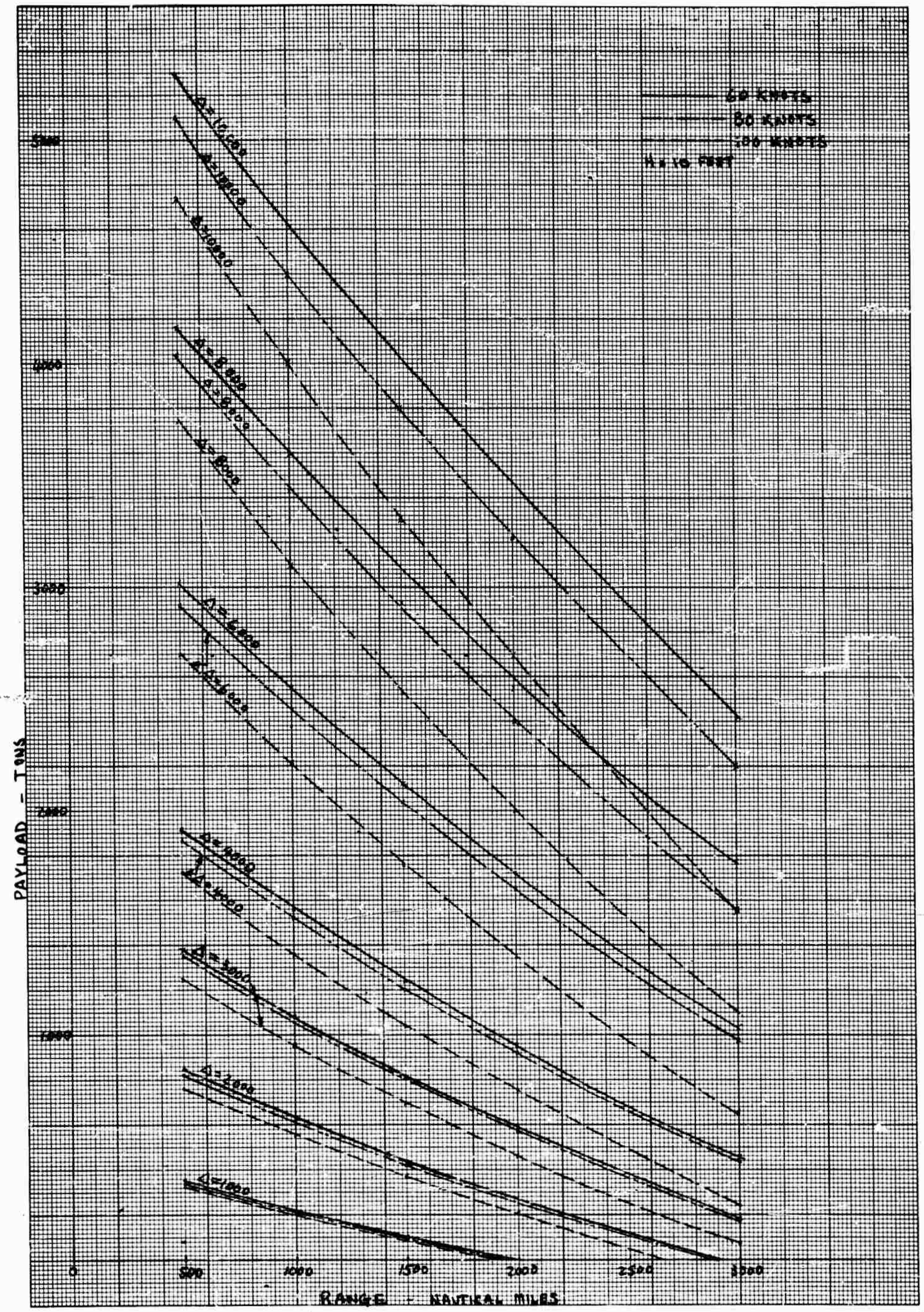


FIG. D-39 GEMS
PAYLOAD VERSUS RANGE, GAS TURBINE POWER PLANT
(Operating Height, 15 Feet;
Speed, 60, 80, and 100 Knots)

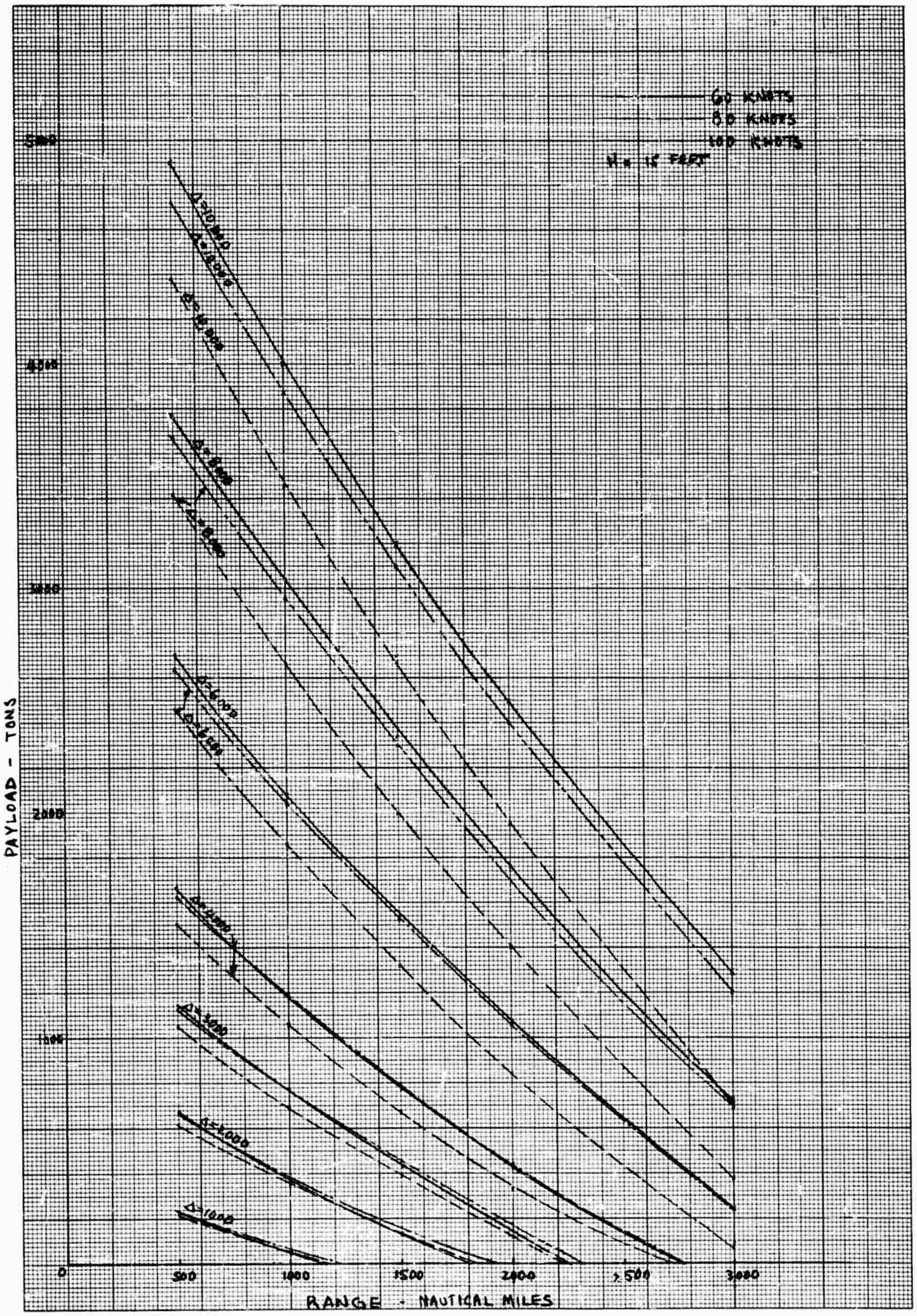


FIG. D-40 GEMS

PAYLOAD VERSUS RANGE, GAS TURBINE POWER PLANT

(Operating Height, 20 Feet;

Speed, 60, 80, and 100 Knots)

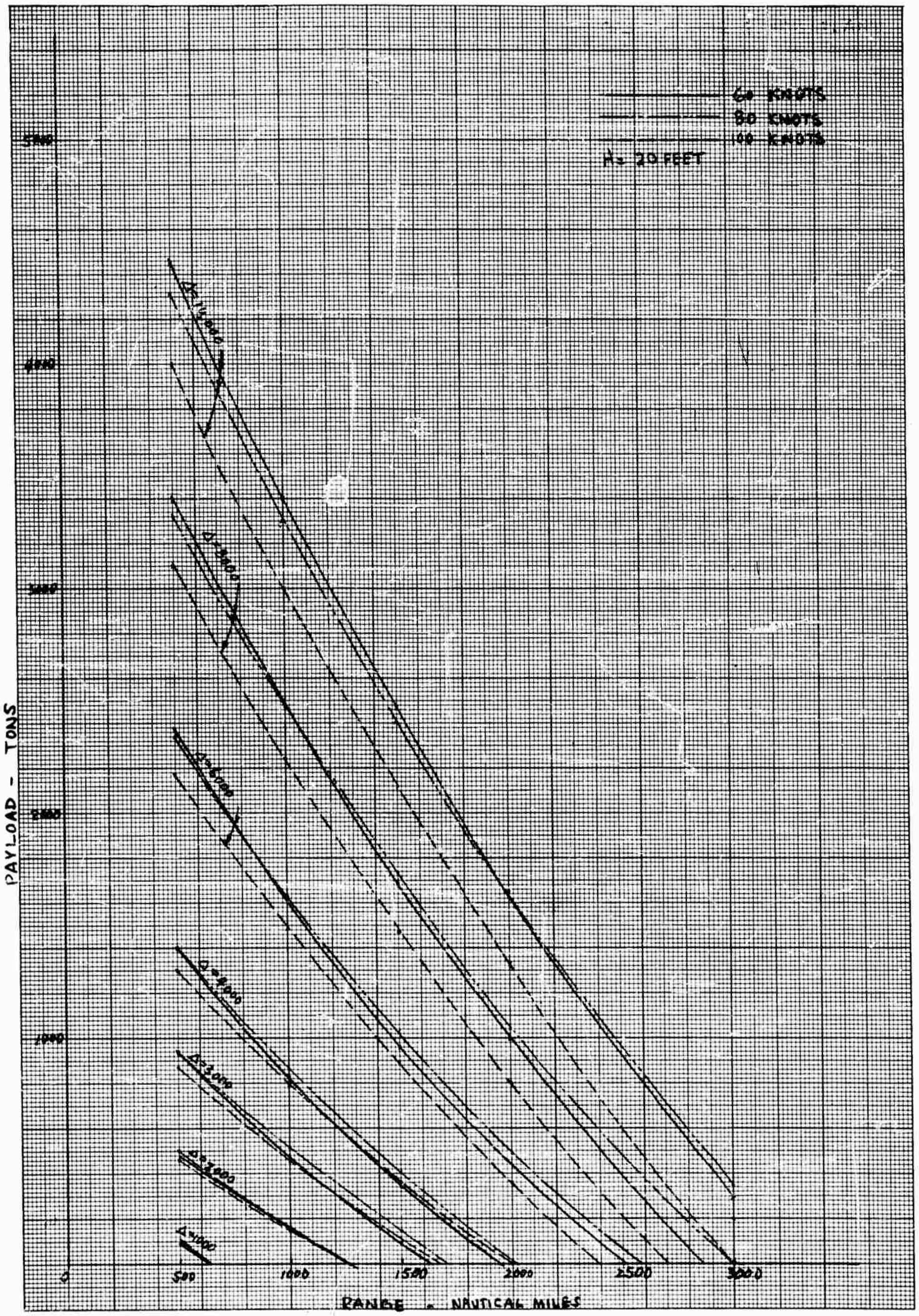
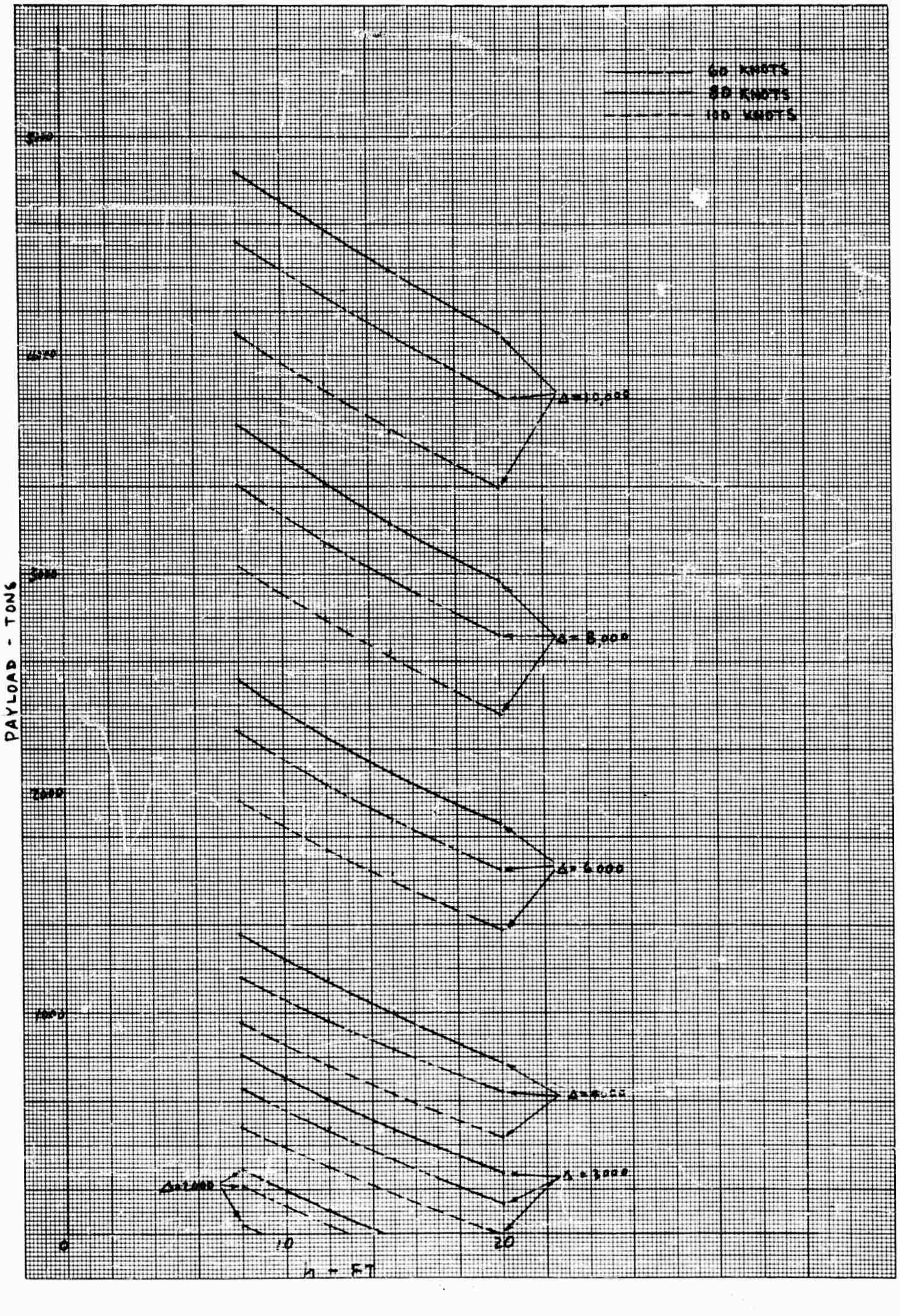


FIG. D-41 GEMS
PAYLOAD VERSUS OPERATING HEIGHT, NUCLEAR POWER PLANT
(Displacement, 2,000 to 10,000 Tons;
Speed, 60, 80, and 100 Knots)



Appendix E

CONTENTS

ILLUSTRATIONS

Illustrations (concluded)

Appendix E

SUBMARINES

Appendix E

SUBMARINES

Introduction

Submarines ranging in payload from 1,000 to 40,000 tons and in speeds from 20 to over 40 knots have been investigated. High performance submarines with payloads of less than 1,000 tons could also be employed in amphibious fleet operations, particularly when acting as convoy escorts. However, submarines in this weight class have not been investigated.

Nuclear power is the only type of power for under-water propulsion on which the submarines of interest can depend. Two types of reactor systems have been investigated: One is the present-day pressurized water system and the other is an advanced system.

A ship running under water with sufficient submergence eliminates wave-making resistance. Submerging a vessel, however, increases its wetted surface and, at low speeds, the resultant increase in frictional resistance more than offsets the gain in wave resistance. As speed increases, the submarine will offer a decided advantage over the displacement ship on a resistance basis. Another advantage of submarines is that they are not affected by bad weather. The advantages of submarines in regard to their potential ability to avoid detection are obvious.

There are many features on the debit side of the ledger. A submarine must resist water pressure and, based on structural considerations, the submarine will have a greater displacement than a surface vessel with the same payload.

Hulls having a body of revolution shape have a minimum resistance form. At large displacements, however, the diameter of these submarines will create severe operational difficulties in loading, unloading, and drydocking. An alternative is to abandon the low-drag form and resort to a rectangular form whereby the volume is obtained by maintaining a given maximum draft and increasing beam and length.

Submarine design and operational problems are considerably affected by the nature and density of the contemplated cargoes. The lower the cargo density, the larger the submarine must be. Dry cargo must be contained in a pressure hull and loaded and unloaded through large openings in the hull. The arrangement and structural design of a dry-cargo submarine is an extremely complicated problem.

These problems are very much reduced if only liquid cargo is carried. Since the cargo tanks can be pressure-equalized, only a minimum pressure hull for living and machinery spaces and for variable cargo tanks need be provided. The cargo can be pumped in and out without introducing structural discontinuities.

As the size and speed of submarines with rectangular cross section increase, the dimensions and power requirements will become excessive. At this point, submarines with circular cross sections and parallel midbodies should be used. The large drafts, however, will probably require offshore loading and discharging of cargo.

Calculation of Shaft Horsepower

Rectangular Cross Section

The horsepowers required for various displacements and speeds were obtained from Reference No. 1, cited below. This reference, however, does not investigate submarines of less than 20,000 tons of payload. To extrapolate to smaller sizes, and to help construct the horsepower curves, the horsepowers were assumed to vary as follows:

$$SHP = KD^{2/3}V^3$$

where

SHP = shaft horsepower

D = submerged displacement = 1.1 times surface displacement

V = maximum submerged speed in knots

K = constant depending on proportions, and appendages, of submarine.

An examination of the data of Reference No. 1 indicates that a value of K of 0.0032 is representative, and this value has been used in this study.

---

1. Russo, V.L., H. Turner, and F.W. Wood, Submarine Tankers, SNAME paper, November 1960.

Circular Cross Section

As for designs using a rectangular cross section, the horsepowers required for various displacements and speeds were obtained from Reference No. 1. This reference, however, does not investigate submarines of less than 10,000 tons of payload. To extrapolate to smaller sizes, and to help construct the horsepower curves, the horsepowers were assumed to follow the same equation as noted above, except that a representative value for K seems to be 0.0022.

Weights and Volumes

There is no real need for these submarines to operate at a depth greater than that necessary to eliminate wave-making resistance. The pressure hull should be designed for a somewhat greater depth to permit recovery from any momentary loss of control. At these relatively modest depths, volumetric problems will be more severe than weight considerations. To be consistent with the other studies covered in this report, however, summary curves have been plotted on a weight basis.

Figures E-1 through E-14 have been prepared on the basis of present-day reactor systems and on the basis of an advanced reactor system which is assumed to require only 50 percent of the weight and space of the present-day systems. A gas-cooled reactor system offering savings in weight and space of this magnitude would appear to be readily feasible in the time period of concern here. Such improvements are conservative as compared to the power plant weight calculations used elsewhere in this report. The over-all results, however, will not be significantly affected, since the propulsion plant weight is not a major percentage of the total for submarines under consideration.

Tanker Configuration - Rectangular Cross Section

A typical cross section taken from Reference No. 1 would include a circular cross section pressure hull within the larger rectangular cross section outer hull. The area between the hulls would be used for liquid cargo. A draft of 36 feet has been assumed to be the greatest acceptable draft with present channel limitations and pier facilities.

Tanker Configuration - Circular Cross Section

A typical cross section taken from Reference No. 1 would include two circular hulls; the inner hull would be a pressure hull for machinery and crew and the outer hull would provide the main cargo tank between hulls. The maximum diameter has been limited to 80 feet to facilitate drydocking and fabrication.

Dry-Cargo Ship Configuration

Although these ships have been labeled as dry-cargo ships, the results will apply approximately to all nontanker configurations. The dry-cargo configuration creates many more operational and structural problems than the tanker configuration. It is assumed that the payload of the dry-cargo configuration for the same surface displacement will only be about 50 percent of the tanker version. This decrease is due to increased structure complexity and outfitting and the increase in volume required for variable ballast tanks. The variable tanks are used to compensate for the variations in cargo densities. However, as cargo is unloaded, some of the cargo spaces will have to be flooded to make up for the lost cargo weight.

Comparison of Summary Curves for Circular and Rectangular Cross Sections of Submarines

For the same speed, payload, and type of cargo (tanker versus dry cargo configuration) a submarine with a circular cross section should have less displacement than a submarine with a rectangular cross section. This is due to the smaller horsepower requirements of the circular cross section configuration. As has been indicated, however, the circular cross section would not be well suited to certain configurations because of the inefficiency in internal hull layout.

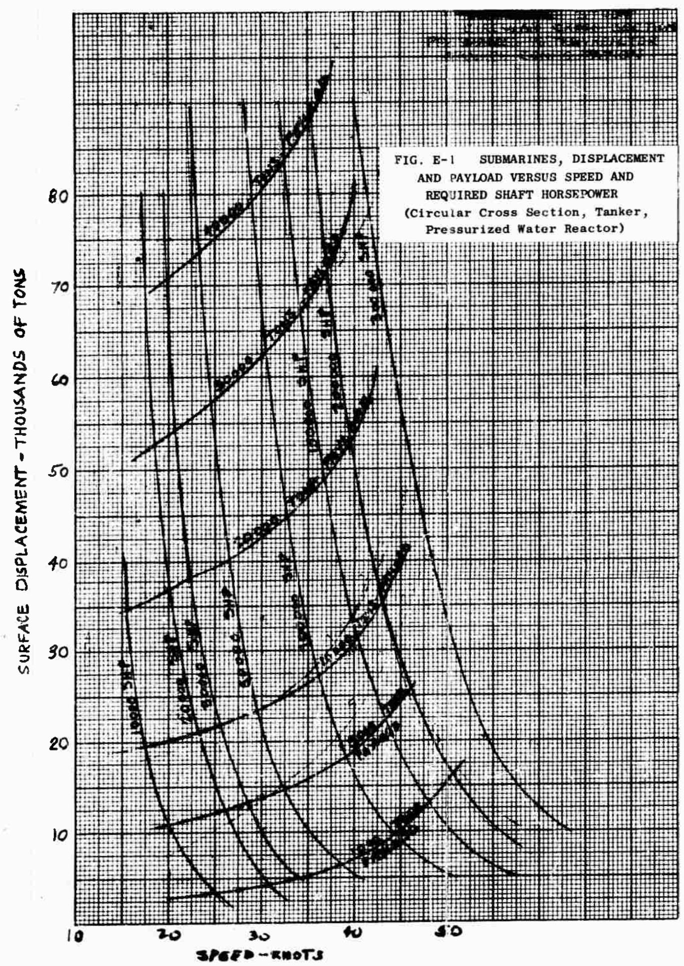


FIG. E-1 SUBMARINES, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER (Circular Cross Section, Tanker, Pressurized Water Reactor)

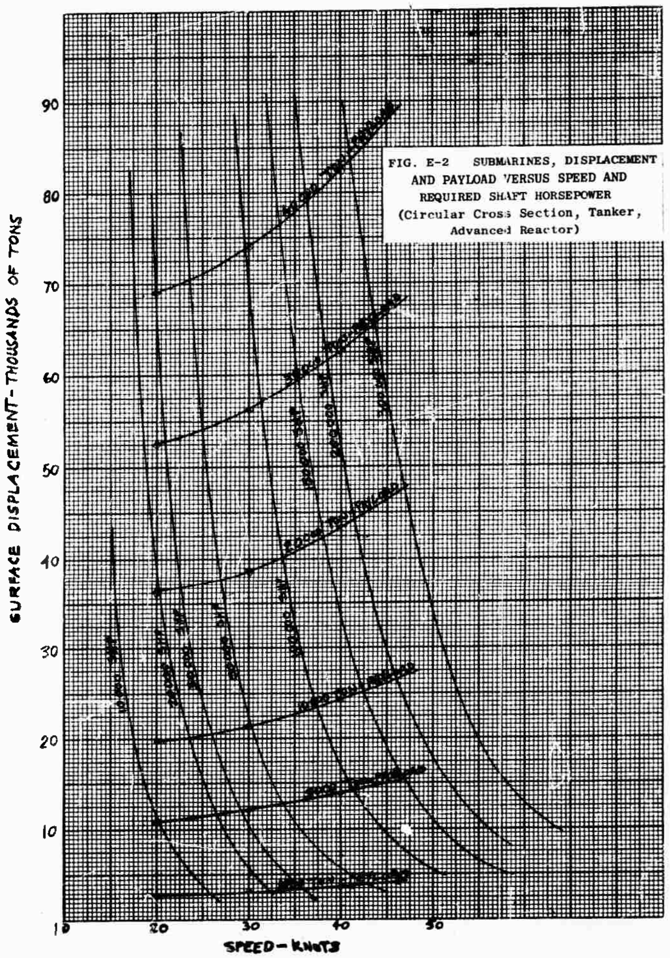


FIG. E-2 SUBMARINES, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER (Circular Cross Section, Tanker, Advanced Reactor)

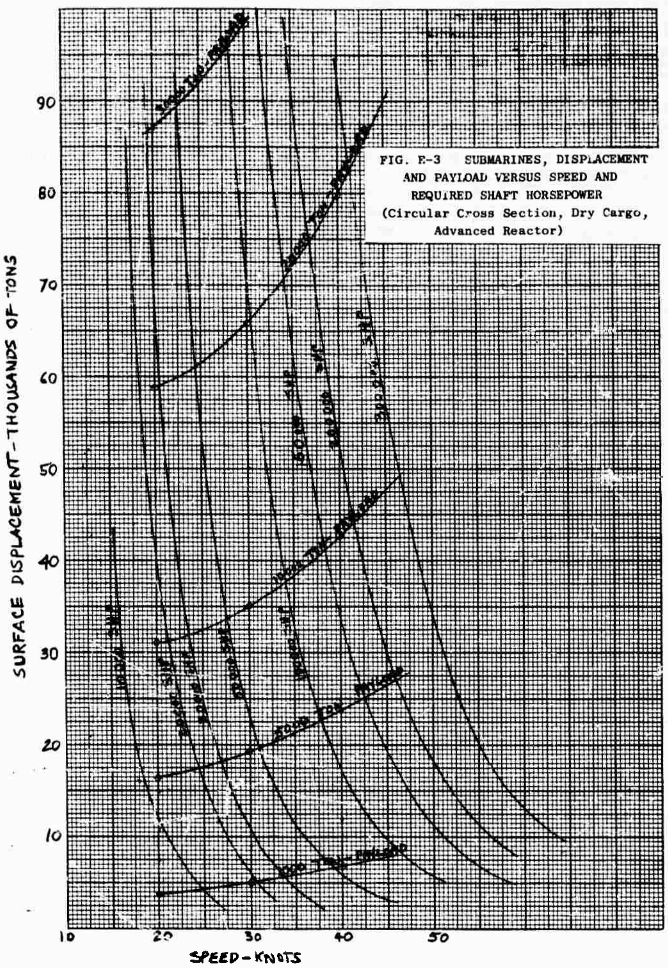


FIG. E-3 SUBMARINES, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER (Circular Cross Section, Dry Cargo, Advanced Reactor)

FIG. E-4 SUBMARINES
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS
AS PERCENTAGE OF TOTAL DISPLACEMENT VERSUS DISPLACEMENT
(Circular Cross Section, Tanker, Pressurized Water Reactor)

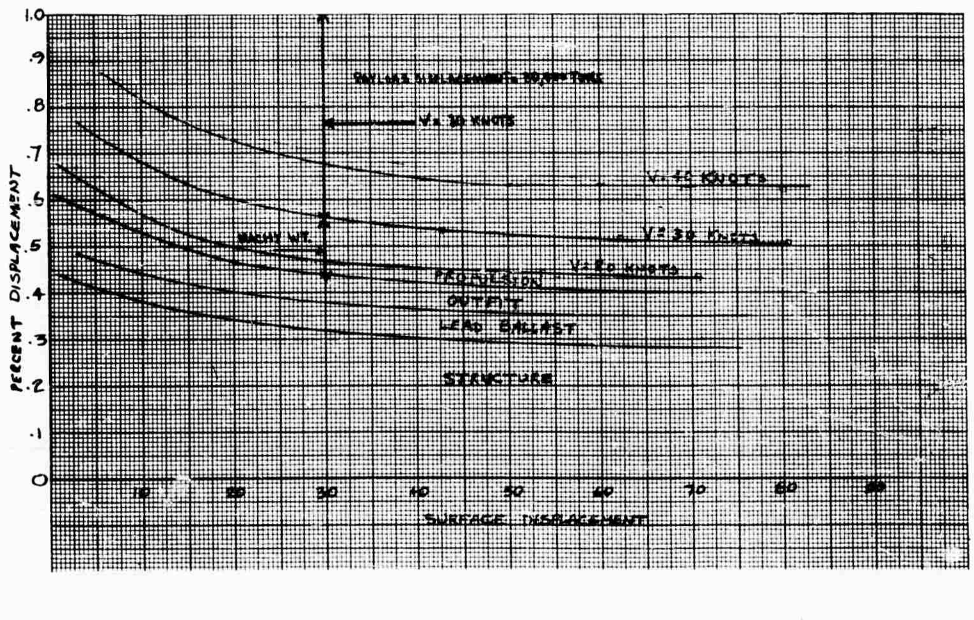


FIG. E-5 SUBMARINES
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS
AS PERCENTAGE OF TOTAL DISPLACEMENT VERSUS DISPLACEMENT
(Circular Cross Section, Tanker, Advanced Reactor)

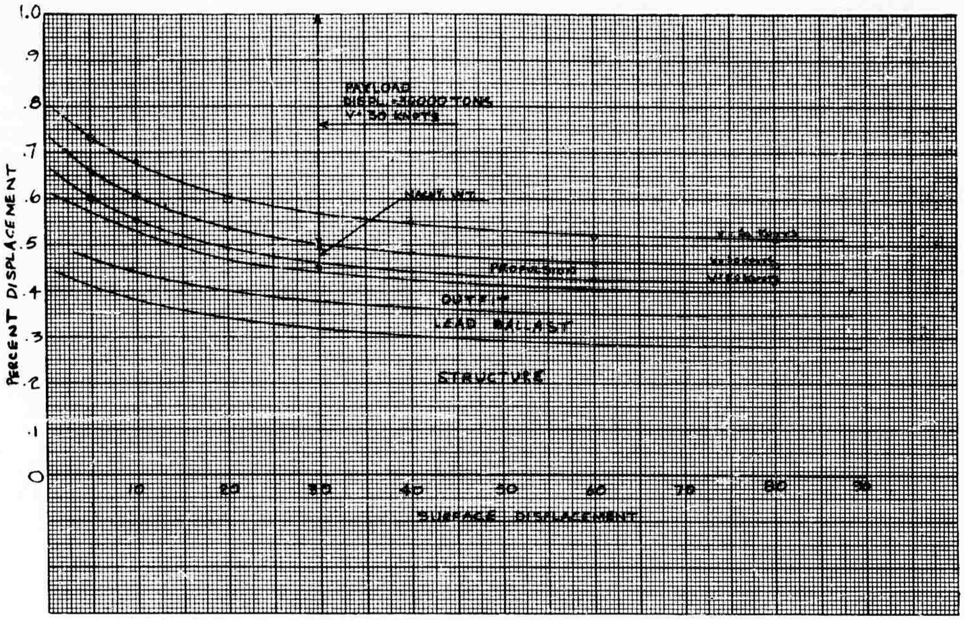
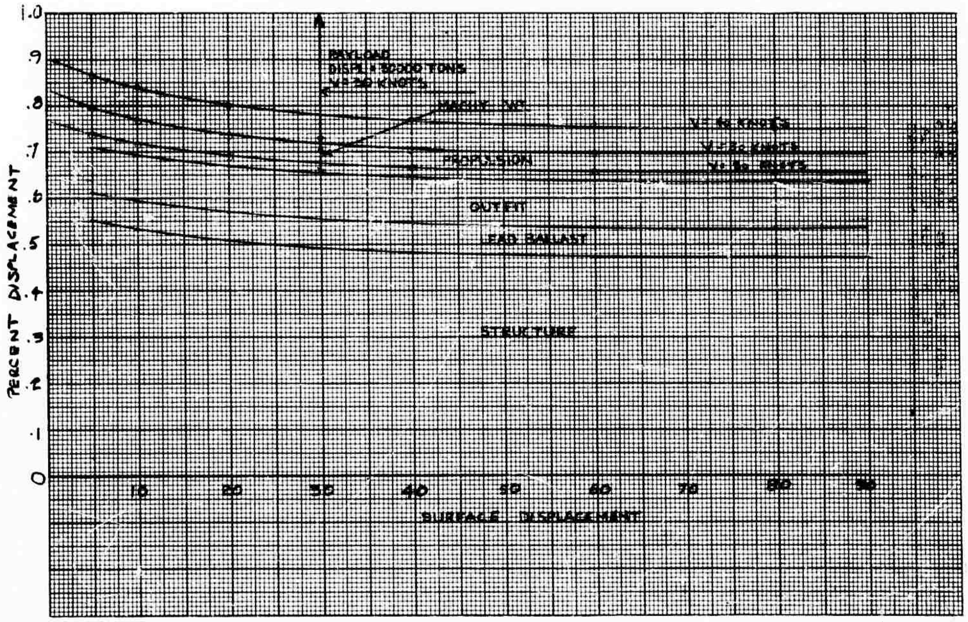


FIG. E-6 SUBMARINES
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS
AS PERCENTAGE OF TOTAL DISPLACEMENT VERSUS DISPLACEMENT
(Circular Cross Section, Dry Cargo, Advanced Reactor)



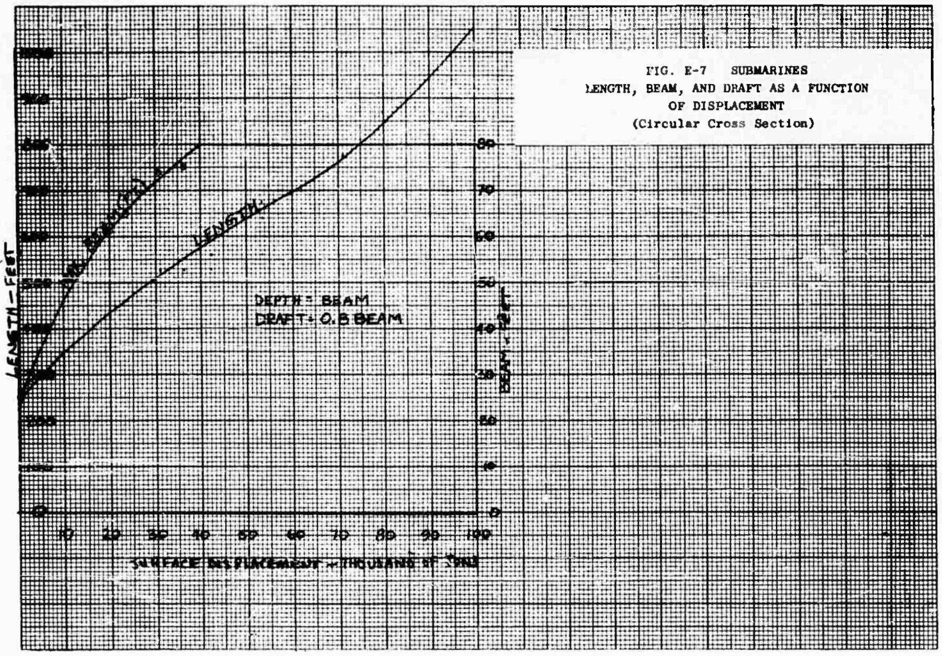


FIG. E-7 SUBMARINES
LENGTH, BEAM, AND DRAFT AS A FUNCTION OF DISPLACEMENT
(Circular Cross Section)

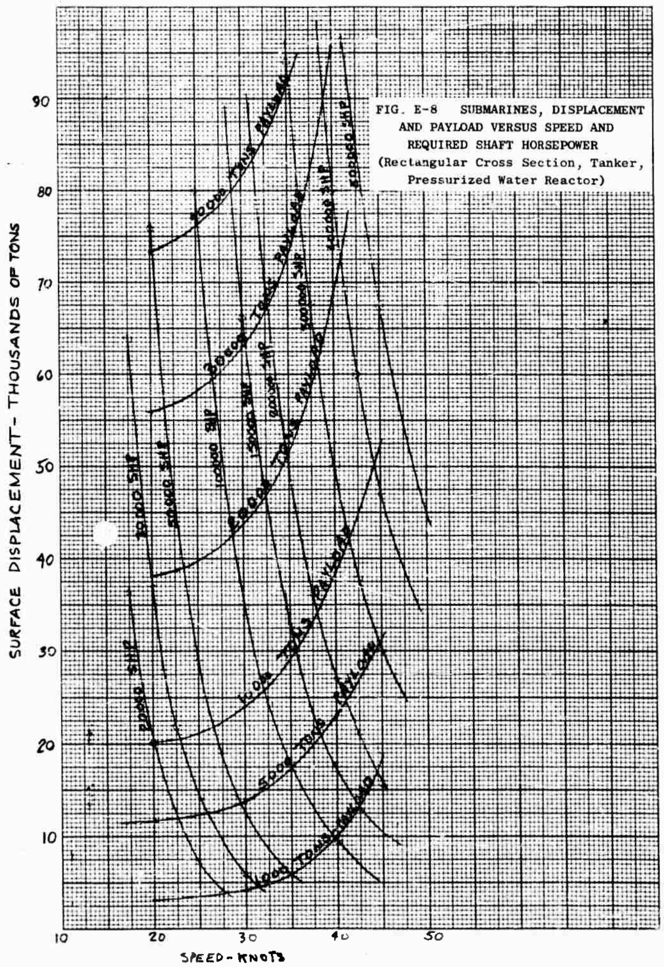


FIG. E-8 SUBMARINES, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER
(Rectangular Cross Section, Tanker, Pressurized Water Reactor)

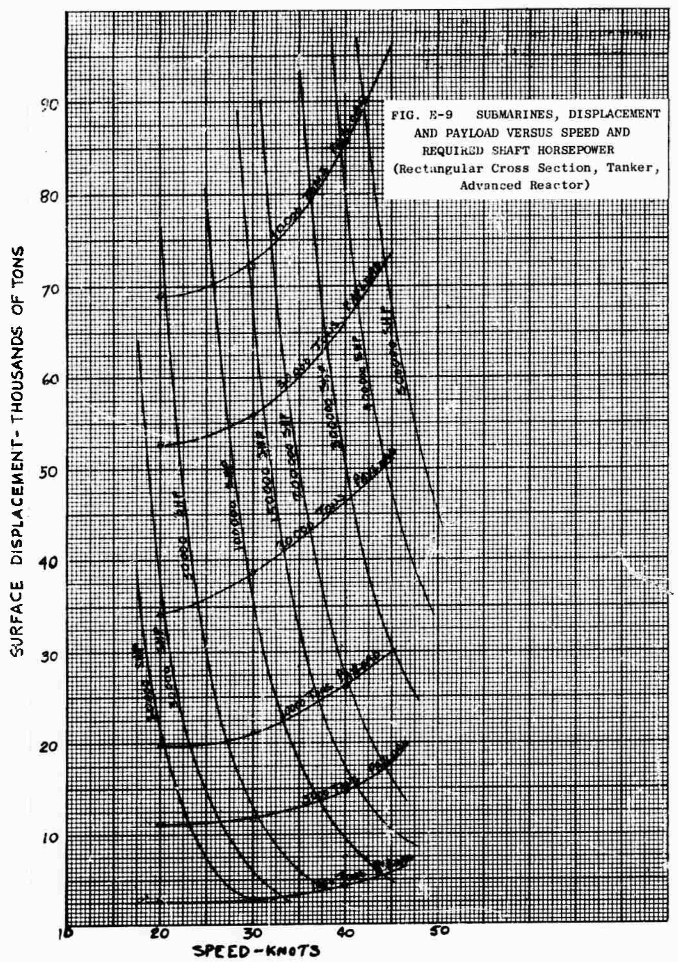


FIG. E-9 SUBMARINES, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER (Rectangular Cross Section, Tanker, Advanced Reactor)

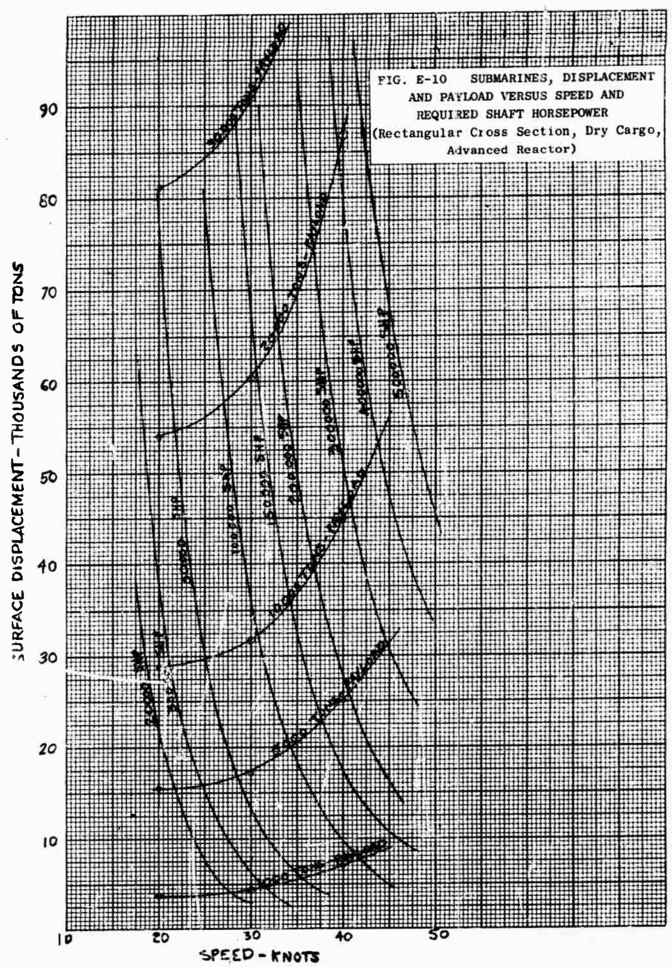


FIG. E-10 SUBMARINES, DISPLACEMENT AND PAYLOAD VERSUS SPEED AND REQUIRED SHAFT HORSEPOWER (Rectangular Cross Section, Dry Cargo, Advanced Reactor)

FIG. E-11 SUBMARINES
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE
OF TOTAL DISPLACEMENT VERSUS DISPLACEMENT
(Rectangular Cross Section, Tanker, Pressurized Water Reactor)

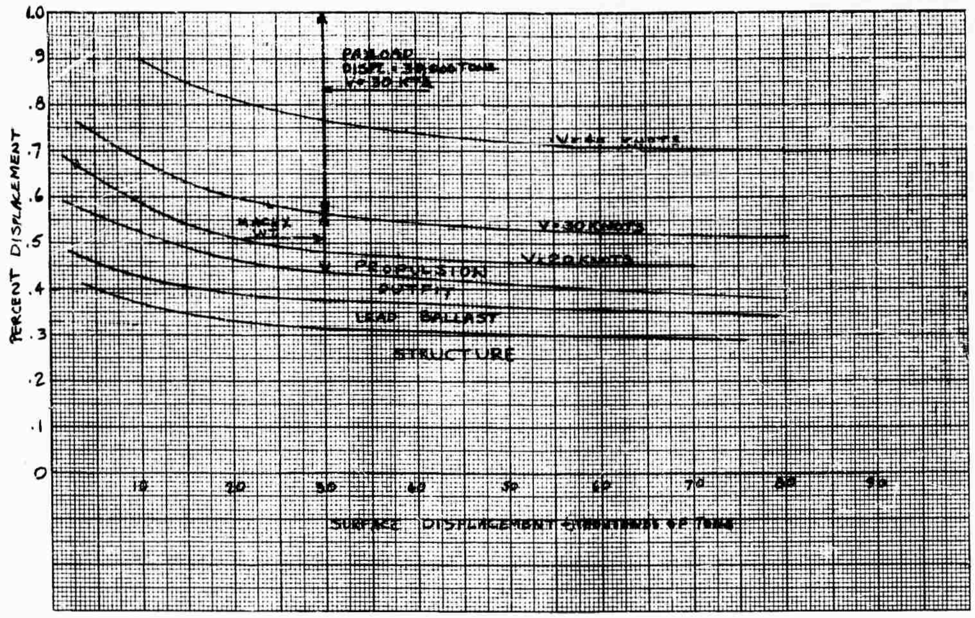


FIG. E-12 SUBMARINES
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE
OF TOTAL DISPLACEMENT VERSUS DISPLACEMENT
(Rectangular Cross Section, Tanker, Advanced Reactor)

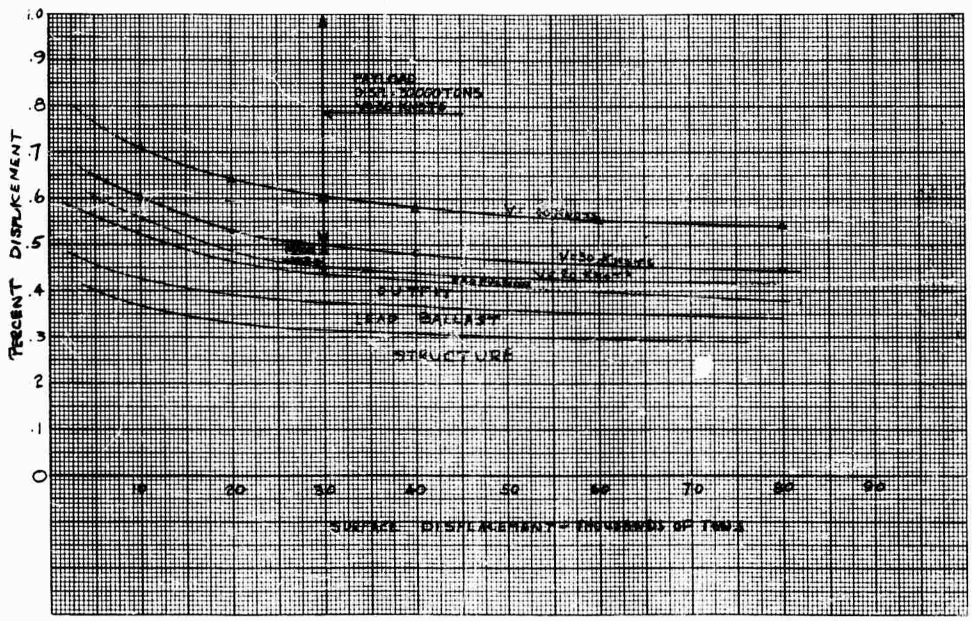
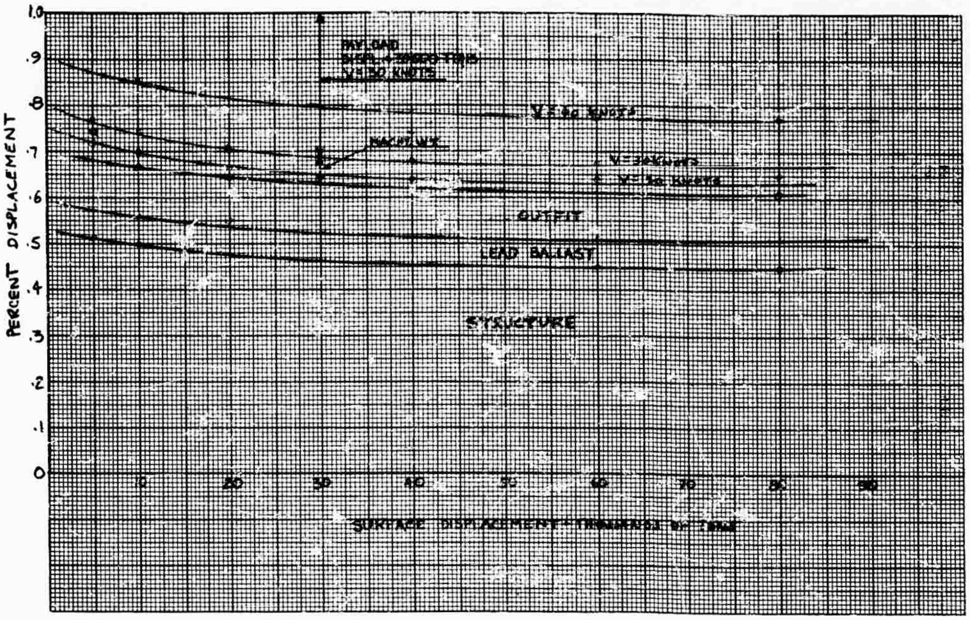


FIG. E-13 SUBMARINES
PAYLOAD POTENTIAL AND OTHER MAJOR WEIGHT COMPONENTS AS PERCENTAGE
OF TOTAL DISPLACEMENT VERSUS DISPLACEMENT
(Rectangular Cross Section, Dry Cargo, Advanced Reactor)



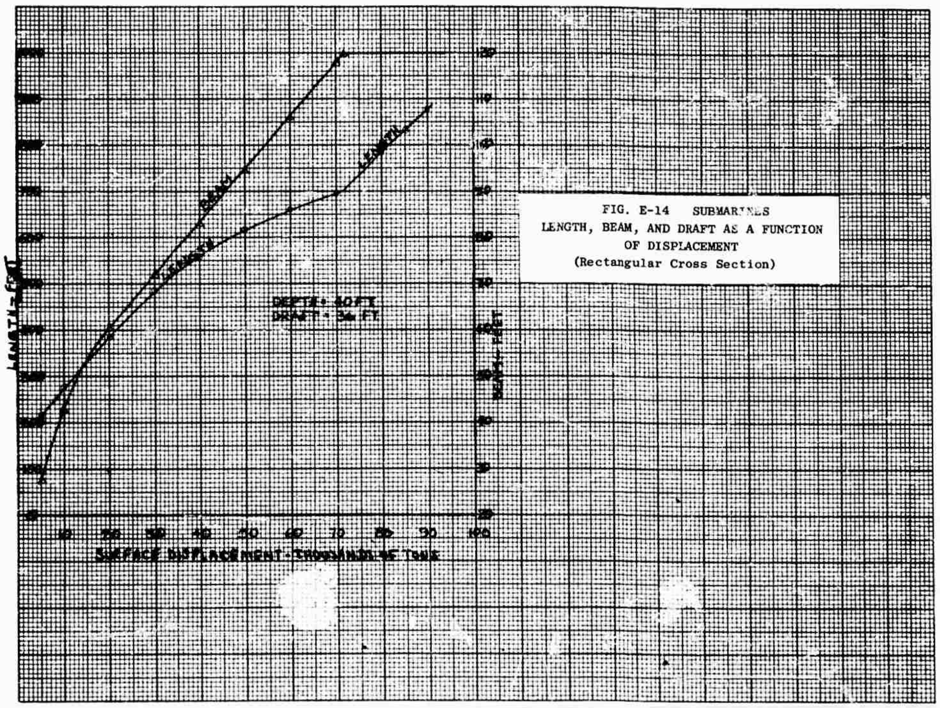


FIG. E-14 SUBMARINES
LENGTH, BEAM, AND DRAFT AS A FUNCTION
OF DISPLACEMENT
(Rectangular Cross Section)

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